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Technical Report

No. 13546

IMPROVED RECOVERY VEHICLE (IRV)

M88A1E1 HYDRAULIC SYSTEM ANALYSIS

September 1991

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Warren, Michigan 48397-5000

NOTICES

REPORT DOCUMENTATION PAGE	Form Approved OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	01 Sep 91	Final Mar 90 - Jun 91

4. TITLE AND SUBTITLE

IMPROVED RECOVERY VEHICLE (IRV)
M88A1E1 HYDRAULIC SYSTEM ANALYSIS

5. FUNDING NUMBERS

6. AUTHOR(S)

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7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

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8. PERFORMING ORGANIZATION REPORT NUMBER

13546

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

Commander
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Warren, MI 48397-5000

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

APPROVED FOR PUBLIC RELEASE:
DISTRIBUTION IS UNLIMITED

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

Current M88A1E1 recovery vehicles do not have the required winch, hoist, and tow capabilities to support M1 tank equipped units. M88A1E1 prototypes built by BMY Corp., to meet this requirement, were found to have serious performance deficiencies during Technical Feasibility Testing (TFT), held at Aberdeen Proving Grounds (APG), in 1989. This report documents 1990-91 APG Summer Testing (ST) which resolved those deficiencies. Failure reports developed during TFT were the source upon which a ST plan was developed. Main winch pull capability, hoist block and tackle, and hydraulic system overheating were critical short comings identified in TFT and resolved during ST.

The ST has identified that a hydraulic oil cooler must be installed to meet the requirements of the IRV purchase description. An assessment is presented which describes typical heat gain characteristics, of the M88A1E1 hydraulic system, during vehicle operation. Actual versus theoretical data has been evaluated and used as the basis for sizing and specifying a representative oil cooler

14. SUBJECT TERMS

Improved Recovery Vehicle	Oil cooler	Hydraulic System
Winch	Hook Block	Test Incidents
Hoist	Duty Cycle	Levelwind

15. NUMBER OF PAGES 172

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED	SAR

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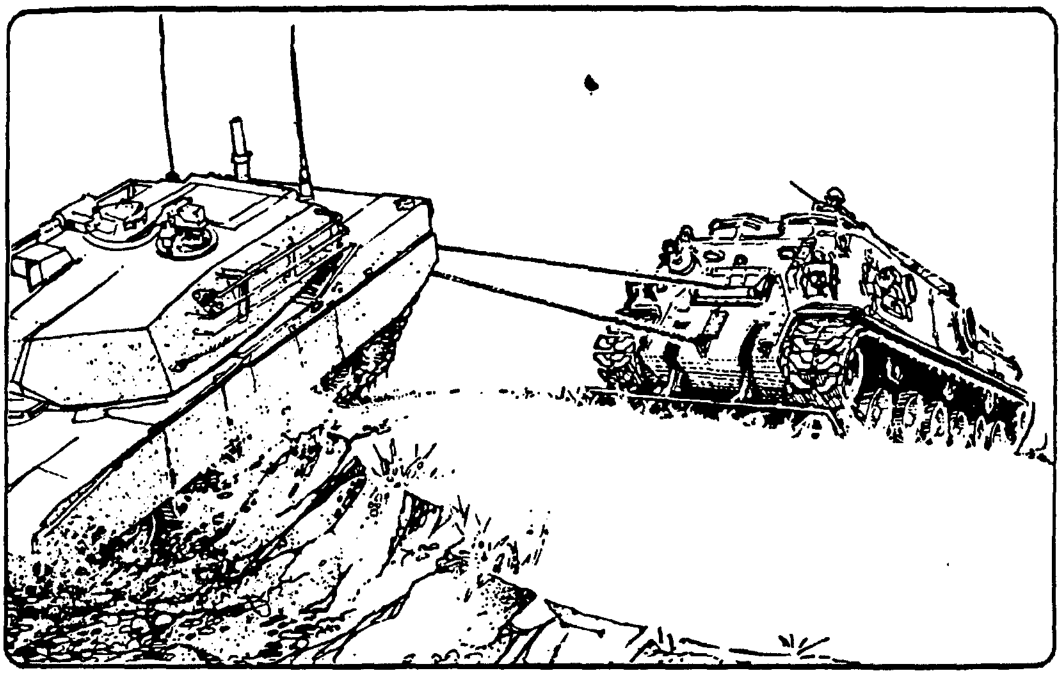
March 1990 thru June 1991



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1.0. INTRODUCTION

The current M88A1 recovery vehicle does not have the required winch, hoist, and tow capabilities to support M1 Abrams Main Battle Tank equipped units. Therefore a need exists for an Improved Recovery Vehicle (IRV). Recovery of disabled tanks is a critical maintenance function. The IRV will enable crews to recover, repair, and return vehicles to the battlefield. The hydraulic system is the heart of the recovery function.

The M88A1E1 (Figure 1), manufactured by BMY Combat Systems, is a medium recovery vehicle intended to replace the M88A1 in the recovery of heavy combat vehicles, to include the M1 series of tanks and future heavy combat vehicles. The M88A1E1, which is a product improved M88A1, has an upgraded powertrain, improved winch, hoist, and tow capabilities, and increased armor protection. It also has an upgraded suspension system and an auxiliary power unit for ancillary tools.

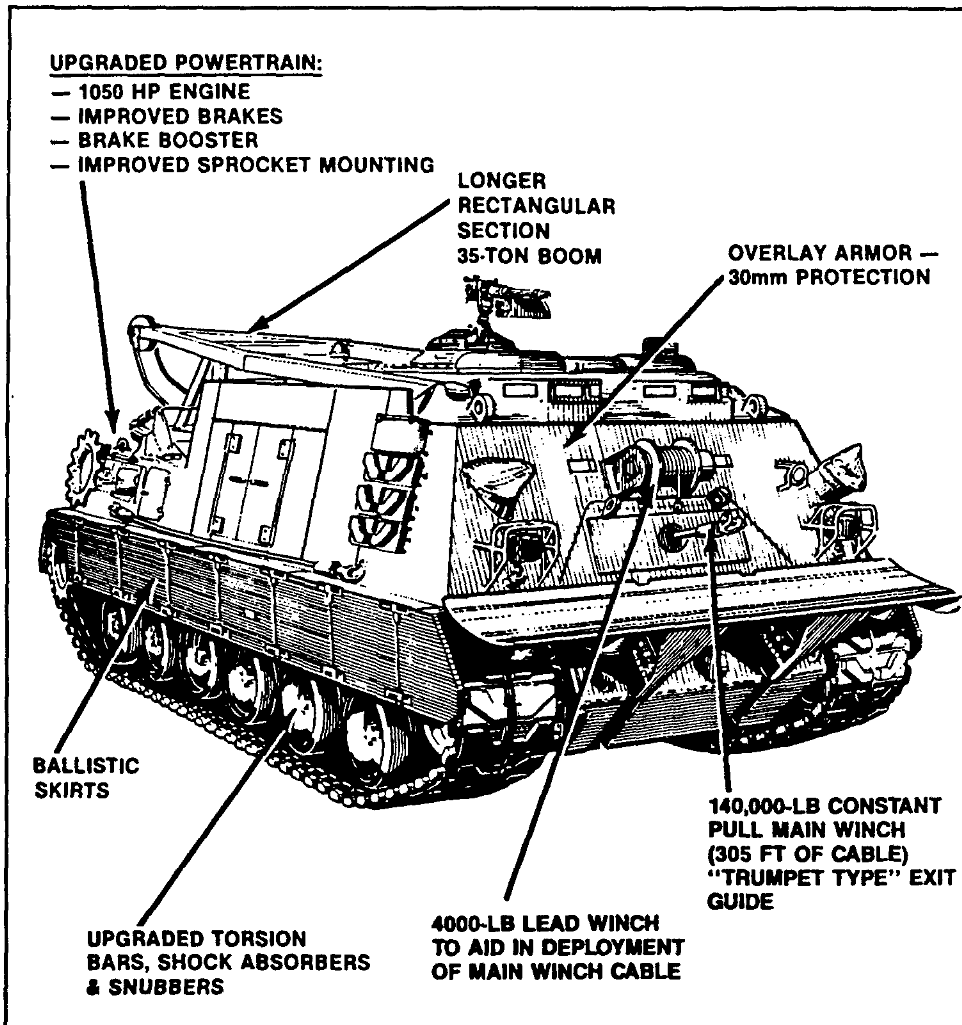

Figure 1- M88A1E1 Improved Recovery Vehicle

According to the performance specification, the IRV must:

a. Lift 35 tons using the spade for stabilization.

b. Remove and replace an M1 series tank turret.

c. Hoist/winch recover a "*nosed-in*" M1-series tank requiring the simultaneous use of the boom and main winch, while using the spade for stabilization.

d. Winch-recover an M1-series tank mired to wheel and fender depth.

e. Use the auxiliary winch to deploy the main winch cable from the IRV out to a distance of 100 meters (m), $\pm$ 10 m, in less than 30 minutes.

f. Rewind the main, hoist and auxiliary winch cables in such a manner and speed as to preclude damage when retrieving under minimal load.

g. Perform four winch/lift cycles in continuous series, per Operation Mode Summary outlined in Test and Evaluation Master Plan.

2.0. OBJECTIVE

The Systems Support Branch (AMSTA-ZDS) of TACOM was tasked to investigate the hydraulic capabilities of the M88A1E1. As a result of the investigation, AMSTA-ZDS would serve as the hydraulic system expert on a team of TACOM professionals developed to assure the U.S. Army is equipped with a vehicle that is "*state of the art*" in recovery systems. To acquire this knowledge, two primary objectives would have to be met:

a. The operational characteristics of BMY Prototype #4 (P4) would be determined through test, and compared to the requirements of the IRV Purchase Description (PD), ATPD 2150, Revision A. System deficiencies would be pinpointed and recommendations as to possible solutions would be developed.

b. Technical Feasibility Tests (TFTs) and PreProduction Tests (PPTs) were performed at the U.S. Army's Aberdeen Proving Grounds (APG) during 1989. Test Incident Reports (TIRs) documenting failures were generated. The AMSTA-ZDS tasking would resolve any failures not reconciled before the PPT was prematurely halted.

3.0. CONCLUSIONS

The M88A1E1 has met the winching and hoisting requirements as stated in the PD. Furthermore, all TIRs from the prior testing were resolved; however, the following shortcomings were noted in testing and must be resolved.

a. During testing, hydraulic oil temperatures as high as 250°F were recorded.

As specified in the PD, reservoir temperatures shall not exceed 170°F. This requirement cannot be met without the addition of a hydraulic oil cooler.

b. The BMY bushing design for the hoist hook block is underrated for continuous operation. The sheave spacing and deadman location is also inadequate. A redesign of the hoisting hook block and tackle, to prevent cable twisting and wear, is required.

4.0. RECOMMENDATIONS

The recommendations provided below, in some cases, have been tested to a limited extent, others have not been tested at all. Those which have been tested are detailed within the body of this report. With respect to limited testing, these recommendations were basically quick fixes that provided satisfactory results in completing the test. These recommendations, if adopted or replaced by other ideas shall be subjected to requirements of the baseline test.

4.1. Hoist

a. The sheave for the hook block should be changed from a bushing to a roller/ball bearing design for continuous operation. Continuous operation with the bushing design generates heat from sliding friction. Heat buildup breaks down lubrication and causes shaft wear.

b. The current location of the boom hoist cable deadman allows the cable to twist during operation. The deadman should be relocated to eliminate this twisting action and to keep cables aligned.

c. Spacing of the boom and hook block sheaves should be equal, as to limit any cable angularity. The fleet angle developed from unequal spacing causes side loading to the sheaves, resulting in bearing and shaft failures.

4.2. Winch

a. A system inefficiency is present within the auxiliary and main winch hydraulic circuit. The current design allows for an excessive buildup of hydraulic pressure which is converted to heat at the main winch. Isolation of the auxiliary winch by the installation of a three-pump system should be investigated to eliminate the inefficiency.

b. A double pilot operated check valve is required in the main winch circuit to lock out the level wind cylinder during no-flow conditions. This valve will eliminate level wind failures caused by cylinder drift.

c. A hydraulic oil cooler should be installed to eliminate excessive temperature buildup. Maximum design conditions (as found in a desert environment) should be used in sizing. These conditions would be based on continuous winching operations held in 120°F ambient air.

d. A fail-safe system should be incorporated in the cable miswrap linkage to prevent main winch housing failure. A "*break-away*" bearing cap for the housing tie-rod should be studied as a possible solution.

e. The level wind mechanism should be reviewed for structural rigidity. Underdesigned parts shall be reengineered.

f. Testing has produced a series of quick fixes (drain and load sense lines, shuttle valve) to the vehicle hydraulic system. These fixes should be reviewed, and where applicable, the fix should be optimized and incorporated into the vehicle design.

g. Accessibility of certain hydraulic system test points are very difficult. The location of all test points should be reassessed. All inaccessible points shall be made accessible where possible, for ease of maintenance and use of diagnostic equipment.

h. Adjustment to main winch motor layer sense unit is required to produce consistent safe stalls on all three drum layers.

5.0. DISCUSSION

5.1. Background

In January of 1987, a research and development sole source contract was awarded to BMY for the development of five prototype M88A1E1s. Based on a run-off between competing contractors, BMY's M88A1E1 was chosen to be the IRV. Their five vehicles were tested at APG from June 1988 to April of 1989. In April of that year, the Secretary of Defense canceled the IRV program due to budget constraints. Approximately seven months later, a Congressional Directive was issued to complete the technical and user testing of the M88A1E1. This directive was issued in Congressional Report 101-345, "Appropriations for the DoD for the Fiscal Year 1990."

In March of 1990, the Tank Recovery Branch (AMSTA-UCB) was tasked by the IRV Program Office to develop a team of TACOM specialists to evaluate the operational status and baseline capabilities of the five BMY prototypes residing at APG. In particular, AMSTA-ZDS was chosen to perform the analysis of the hydraulic systems. Each TACOM team was given a separate prototype to conduct their respective analysis on. Vehicle P4 was chosen for the hydraulics analysis.

During the interim time of program cancellation, a significant amount of test data was lost, and test engineers critical to the program had been assigned to other projects. Therefore it was necessary for AMSTA-ZDS to assemble a group of TACOM and Combat Systems Test Activity (CSTA) engineers and technicians to test and analyze P4's hydraulic systems during the summer of 1990 (August 1990 through March 1991). A list of team members is provided in Appendix A.

At the start of the Summer Test (ST) at APG, little knowledge was available as to the physical condition and actual operation of the five vehicles. Remaining data from the prior test was sorted through, and it was found that the TIRs would piece together the past problems and deficiencies. Appendix B tabulates all of the major TIRs that were found. The table states the subject area and resolution of the incidents. As a result of the ST, all major TIR issues were addressed. Having completed research of the hydraulic functions of the M88A1E1, AMSTA-ZDS drafted a Test Directive (TD) to be implemented by CSTA support staff. The TD is included as Appendix C. The TD was written to test the M88A1E1 for its compliance against the requirements set forth in the PD.

5.2. M88A1E1 Hydraulic Functions

The M88A1E1 is made up of complex hydraulics that operate the critical recovery and maintenance roles. The hydraulic system is divided into three functional areas:

a. Main/Auxiliary Winching

b. Hoist Winching

c. Ancillary Tools

The vehicle hydraulics is designed as a closed-loop hydrostatic drive. Appendix D is a schematic detailing the M88A1E1 hydraulic system. It is a load sense system that detects pressure at the load and sends it back to the pump. Upon being signalled of the actual pressure/loading requirement, the main pump will adjust flow, either up or down, to meet the system requirements. The main system is driven by a variable volume load compensated pump. A charge pump is installed to prime the main pump. The main system pump is the principal unit which drives the hydraulic motors for the main, auxiliary, and hoist winches, and auxiliary power unit. The main system operating pressure is approximately 4,000 pound per square inch (psi), and uses MIL-L-2104D 15W40 oil as the hydraulic fluid.

5.2.1. Main/Auxiliary Winch

The M88A1E1 is comprised of two separate horizontal acting winch systems, the main and auxiliary (aux) winch. The main winch is designed to generate 140,000 $\pm$ 10% pounds force (lbs.) single line pull at any point over the entire operational length of cable. As the winch rotates, the cable is laid on the drum by a level wind mechanism (Figure 2). The mechanism is comprised of a hydraulic cylinder that moves the nose piece trumpet back and forth. In doing so, the cable is wrapped side by side across the width of the drum before starting a new layer. The cylinder is controlled by a directional valve that senses the position of a cam which tells the cylinder the direction to move. Additionally, the cam is interfaced with a follower by means of a bracket (Figure 3). The follower rides on a diamond screw that is timed to the rotational speed of the drum (Figure 4).

When cable is wrapped from one layer to the next, increased torque must be supplied. To achieve this, the main winch housing has mounted to it a cable layer sense mechanism comprised of a series of pressure control valves and drum roller follower (Figure 5). As the cable spools onto the main winch (inhauling), the layer sense roller determines the present cable layer and adjusts winch torque accordingly. Layer determination is accomplished through a load sense operation. Load sense ports are marked with an "X" (X-port) and located on the layer sense valve and winch motor. The X-port on the winch reads zero psi for the bare drum (first layer). No volume compensation is needed, since minimum torque is required. As the second layer begins to wrap, the torque requirement increases. Since the same system pressure is acting over a larger moment arm, increased torque is achieved by increasing motor displacement. In this case, the X-port pressure increases to 70 psi

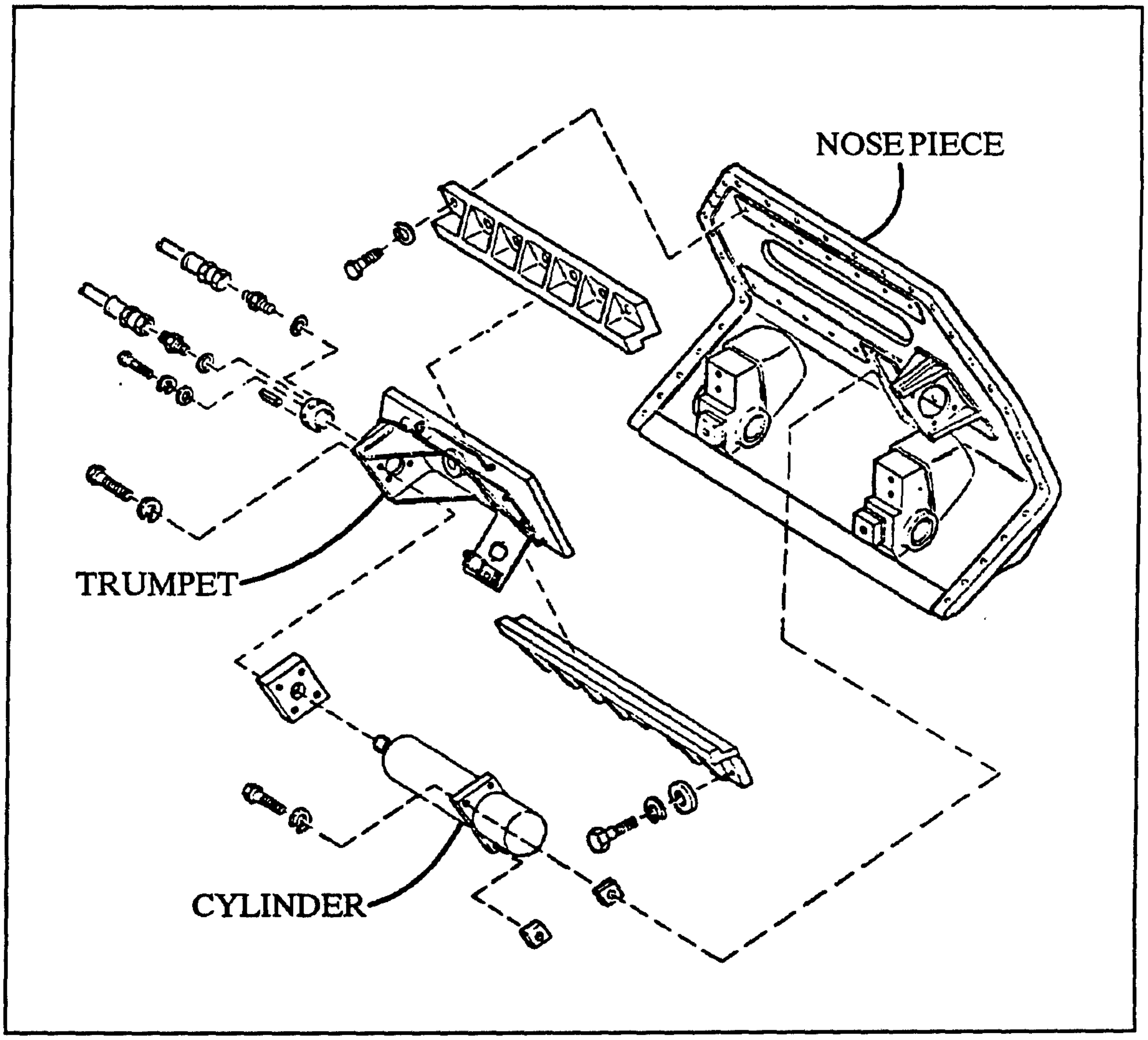

Figure 2 - Level Wind Mechanism (Partial A)

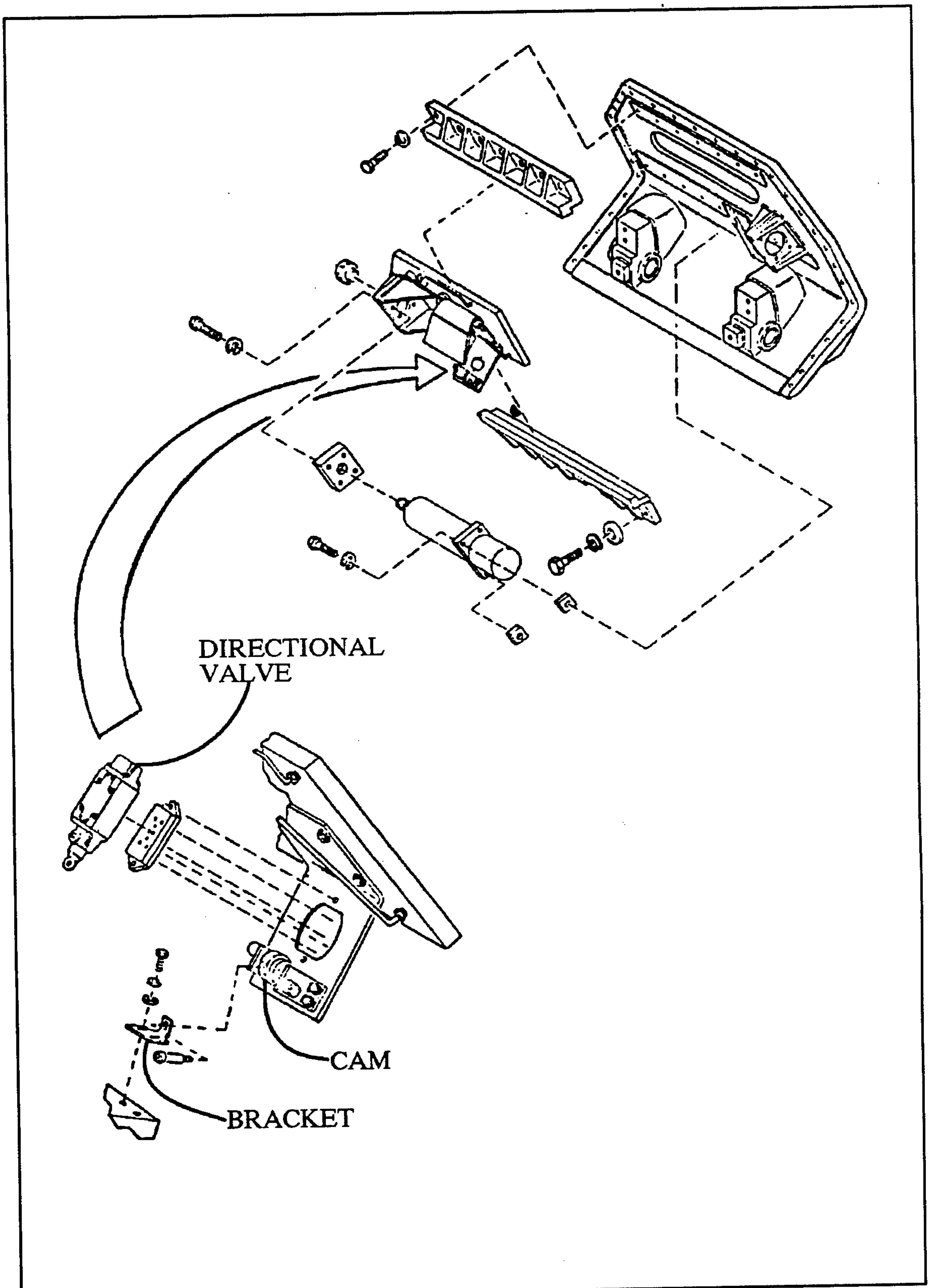

Figure 3 - Level Wind Mechanism (Partial B)

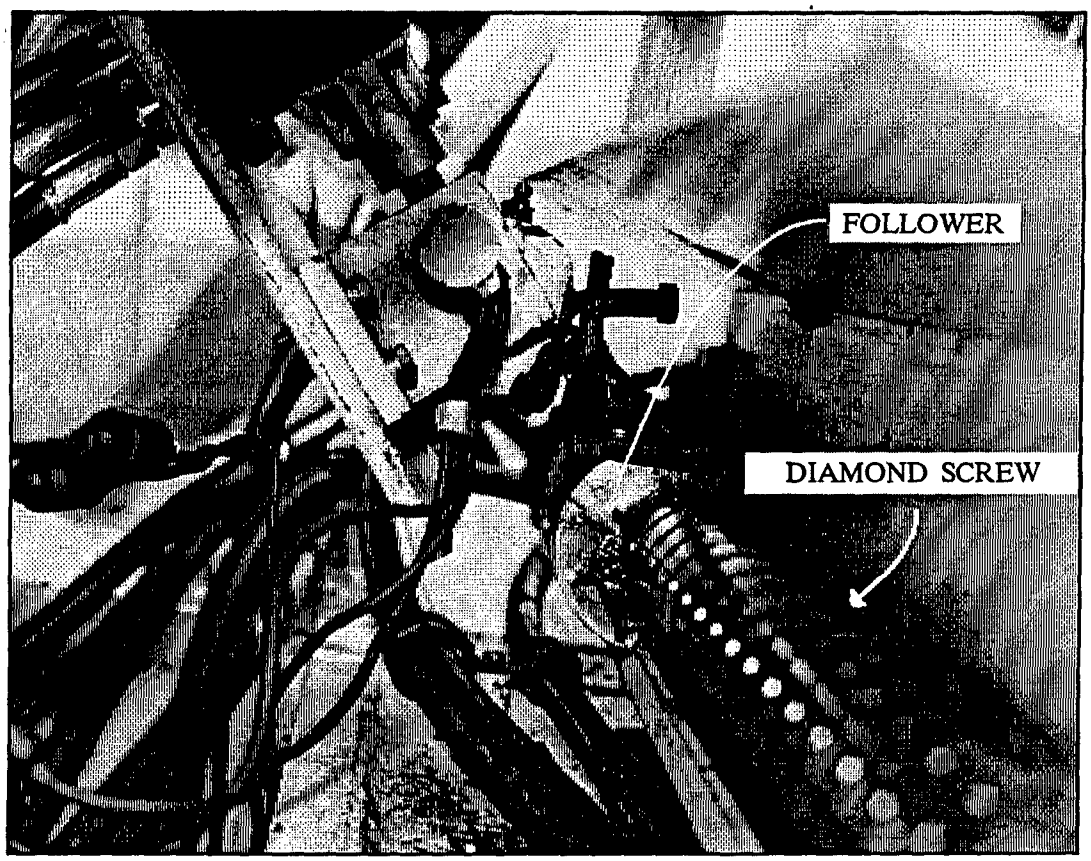

Figure 4 - Level Wind Mechanism (Actual)

which is sensed by the swash plate in the winch motor. The swash plate adjusts to increase its volume output and therefore elevates the motor torque. The winch speed increases from 6.7 feet per minute (fpm) on the bare drum to 7 fpm on the second layer. For the third layer, the X-port pressure increases to 310 psi, and the winch speed increases to 9 fpm.

An aux winch is the second winch provided with the vehicle. It is used to deploy the main winch cable with the assistance of one soldier. The aux winch furnishes power required to outhaul the main winch cable for a recovery operation.

5.2.2. Hoist Winch

The hoist winch is designed to lift 70,000 $\pm$ 10% lbs. to a hook height of 19.7 feet (ft.) at 8 ft. from the front of the vehicle, for not less than 30 minutes on level terrain. The winch reels in cable at a rate of 30 fpm under loads sufficient to allow for proper spooling. The boom and hook hydraulically raise from the travel lock position to a height of 19.7 ft. above ground level with an 8 ft. nominal reach in front of the hull in 90 seconds. The boom returns to the travel lock position in 100 seconds. With the spade emplaced, the boom is capable of moving a 70,000 lb. vertical hanging load through a fore and aft distance of 4 ft. The hydraulic relief valve controlling live-boom movement is set such that with the boom fully extended, no live-boom movement occurs for vertical loads of 75,000 lbs or more.

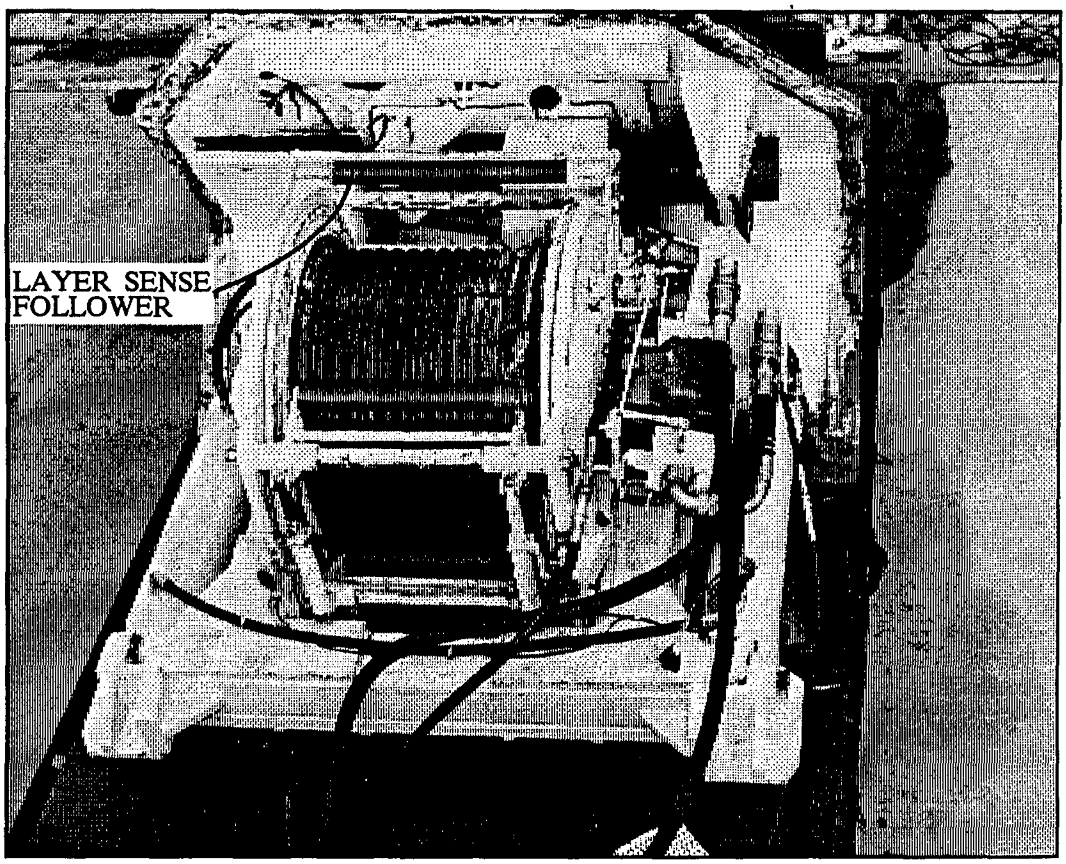

Figure 5 - Layer Sense Mechanism

5.2.3. Ancillary Tools

Ancillary tools fall under the scope of the auxiliary hydraulic system. This system operates through an auxiliary power unit comprised of a 10.8-horsepower diesel engine, 500 psi hydraulic pump, and 28-volt electrical generator. Under no-load conditions, the auxiliary system has sufficient power to operate all ancillary tools for the M88A1E1. Ancillary tools include an impact wrench and fuel transfer system (pump). The fuel transfer pump handles both the refuel and defuel functions. It is capable of transferring fuel from the M88A1E1 to a remote receptacle at a rate of 25 gallons per minute (gpm). The system is also able to transfer fuel from a remote site to the vehicle at a rate of 15 gpm.

5.3. APG Test

Testing of the hydraulic systems took place at APG's Munson Mile Loop Test Course (ML). This course was selected because of the permanent "*deadman*" to anchor the recovery vehicle. Anchoring was necessary, as the vehicle would pull itself forward during a 140,000 lb. main winch pull. The spade (which normally is used for anchorage) was not used, in order that maintenance could be performed more easily and to reduce cable abrasion, since no spare cables were available. Cable abrasion from the level wind trumpet occurs when an angle other than horizontal, between the recovery and disabled vehicles, is encountered. The problem is worse when the vehicle is on its spade on hard terrain.

In order to achieve the desired 140,000 lb. pull, three support vehicles were linked together. The mass to be towed consisted of a train of vehicles including two M88A1E1s and an M60 tank. At the other end of the course, P4 was connected to an M60 tank, which in turn was anchored to the deadman. A load cell was hung from the boom of the lead M88A1E1 towed vehicle. Figure 6 provides a sketch detailing the winching test set-up.

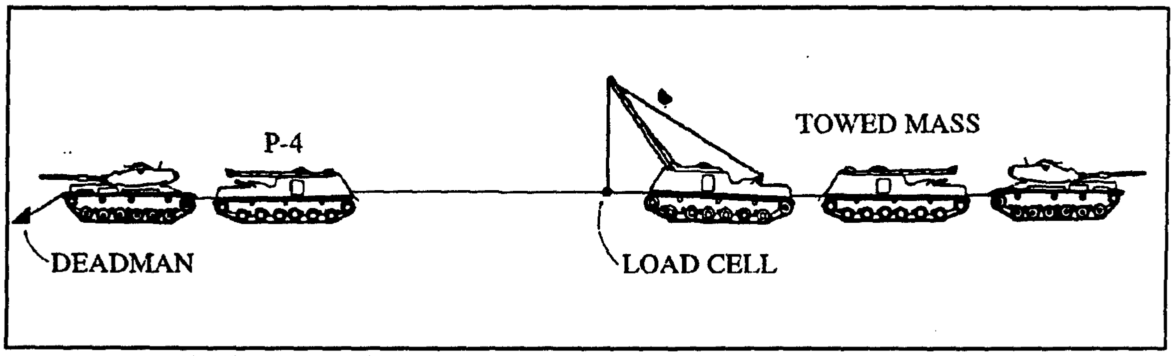

Figure 6 - Winching Configuration

Before rolling out to the test course the vehicles were taken to the lab and fitted with thermocouples, pressure transducers, and data collection instruments. Appendix E gives a complete listing of test instrumentation used in the ST (reference Appendix D schematic for location). The following temperatures and pressures were monitored throughout the test:

a. <u>Temperatures</u>

Reservoir

Dipstick (gear box main winch)

Hoist winch return

Ambient outside air

Crew compartment

Inlet auxiliary winch

Outlet auxiliary winch

Directional control valve hoist raise (HR)

Directional control valve hoist lower (HL)

Directional control valve main input (MI)

Directional control valve main output (MO)

b. Pressures

Charge pump

Main pump compensation

Main pump

X-port main winch

Directional control valve main input (MI)

Directional control valve main output (MO)

With all the instrumentation and measurement equipment in place, an initialization and check of the test equipment commenced. After ten minutes of engine idle, a hydraulic line used in the measurement of the main system pressure burst. A full-force hydraulic oil stream struck a TACOM engineer in the chest. Fortunately, the start-up pressure of 1500 psi and oil temperature of 87°F were low enough as to not cause injury. The engineer was taken to the base hospital, treated, and released. The incident was officially reported. Repairs were made to the vehicle, and the instrumentation all functioned properly.

All test vehicles were taken to the ML, and the team officially began the ST on 15 Aug 90. Appendix F gives a chronology of events that occurred during the ST by date. Appendix G presents a detailed table listing major ST incidents and their respective resolutions. The ST was broken up into four sections; stall, winch, hoist, and duty-cycle tests. Appendix H presents the detailed raw data accumulated for the eighty-one test runs completed. The following sections describe each test.

5.3.1. Stall Load Testing

The first array of tests to be executed were stall loads. Stall loading measures the force required to stall (stop rotating) the main winch drum. This is a safety feature of the system. Without it, the winch would continue to increase its pulling force until the fatigue strength of the cable is reached or drum, failure occurs. There are three layers of cable on the main winch drum and the moment arm is increased as each layer of cable is added to the drum. Therefore the winch must produce a variable torque so a constant load can be inhauled. Table 1 lists the stall loads measured.

Table I - Stall Load Data

No. Cable Wraps	Stall Load (lbs.)	System Pressure (psi)
Bare Drum		
1st Trial	173,900	4250
2nd Trial	178,490	4250
3rd Trial	155,500	4328
4th Trial	155,220	4300
5th Trial	155,790	4300
6th Trial	159,730	4300
7th Trial	162,730	4300
2nd Layer		
1st Trial	165,300	4210
2nd Trial	166,490	4225
3rd Trial	159,000	4390
4th Trial	151,680	4250
5th Trial	152,490	4280
6th Trial	157,540	4250
7th Trial	158,390	4250
3rd Layer		
1st Trial	146,890	4200
2nd Trial	150,550	4230
3rd Trial	149,995	4390
4th Trial	142,690	4275
5th Trial	135,850	4280
6th Trial	148,500	4200
7th Trial	150,750	4250

Data from the above table shows a decreasing stall load with each increase in cable wrap layer. This indicates the appropriate torque is not being generated to produce consistent safe stalls over the entire length of the cable. Stall loads should not exceed 154,000 lbs. (10% over the rated load), or else damage could occur to the winch drum. The winch motor displacement should be adjusted to ensure the layer sense mechanism strokes the motor swash plate to an angle that produces sufficient torque to stall the winch at approximately 154,000 lbs. on each layer.

As noted earlier, stall tests were the first tests conducted; however, they did not proceed without their share of problems. During winch pay out, the level wind mechanism broke twice. To be specific, the follower bracket and shoulder bolt interfacing the level wind cam and diamond screw follower failed. The mechanism was disassembled to analyze the failure. The observation revealed a fractured follower bracket, sheared shoulder bolt, and the probability of flexing in the cam shaft. Bending of the cam shaft may have caused the cam to bind while trying to move, consequently shearing the bolt and bracket. After the first failure, the bracket was replaced; however, following the second failure, a gusset was added to the bracket to increase rigidity and strength (Figure 7). The shaft which holds the cam was also strengthened (Figure 8). The original shaft was a soft steel with a Rockwell "C" hardness of 13. The new shaft installed was case hardened to a Rockwell "C" of 45. The level wind break was the first major failure of the ST.

Since the nosepiece needed to be removed from the vehicle to service the winch, CSTA took the opportunity to develop a "*ground hop*" kit to "*pretest*" the level wind mechanism and to preform timing operations. A ground hop kit (Figure 9) is basically a set of extended length hydraulic hoses required to operate the hydraulics in the nosepiece when removed from the vehicle (Figure 10). In this mode, the winch cable can be reinstalled on the bare winch drum and properly timed. This is not possible if the winch is in the vehicle.

With new level wind components installed, winch timing was conducted to ensure proper cable spooling. During the cable rewrap and timing, the two winch safety switches, fleet angle and cable miswrap, were falsely activated to determine and ensure their operability. Both safety features, when activated, set off an audible alarm in the crew compartment signaling the operator to shut down winch power

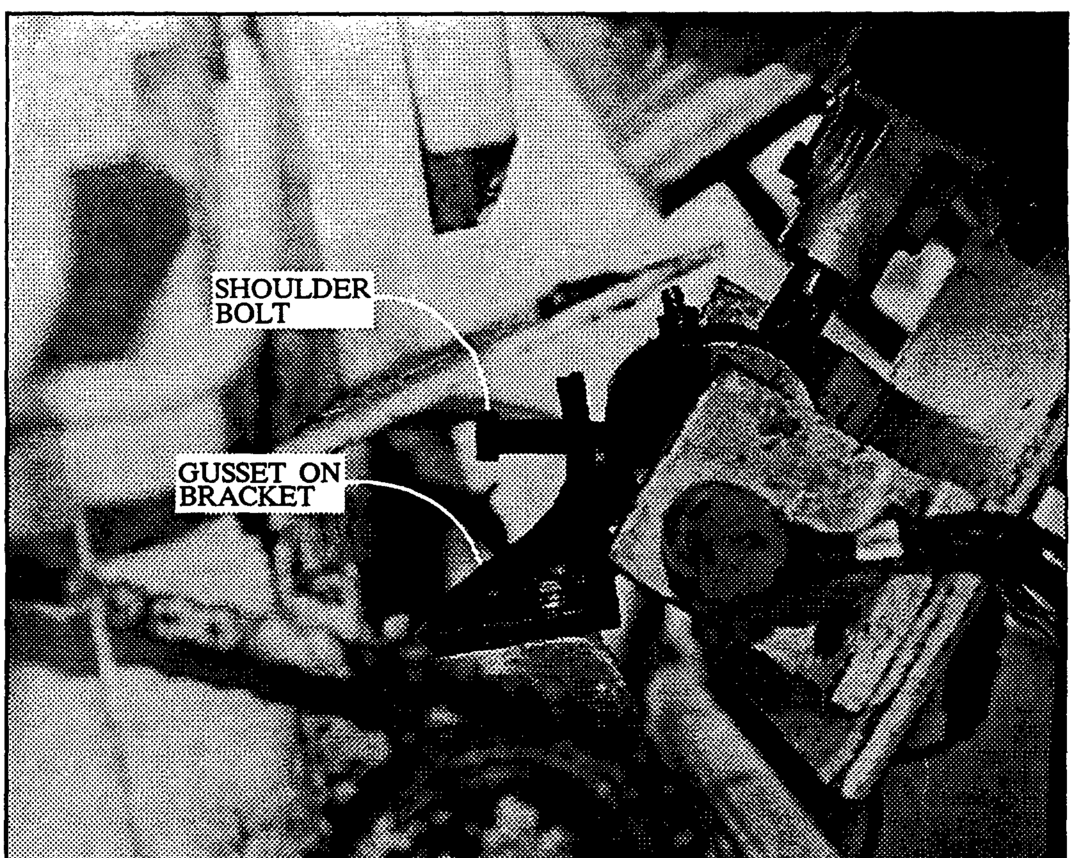

Figure 7 - Strengthened Level Wind Bracket

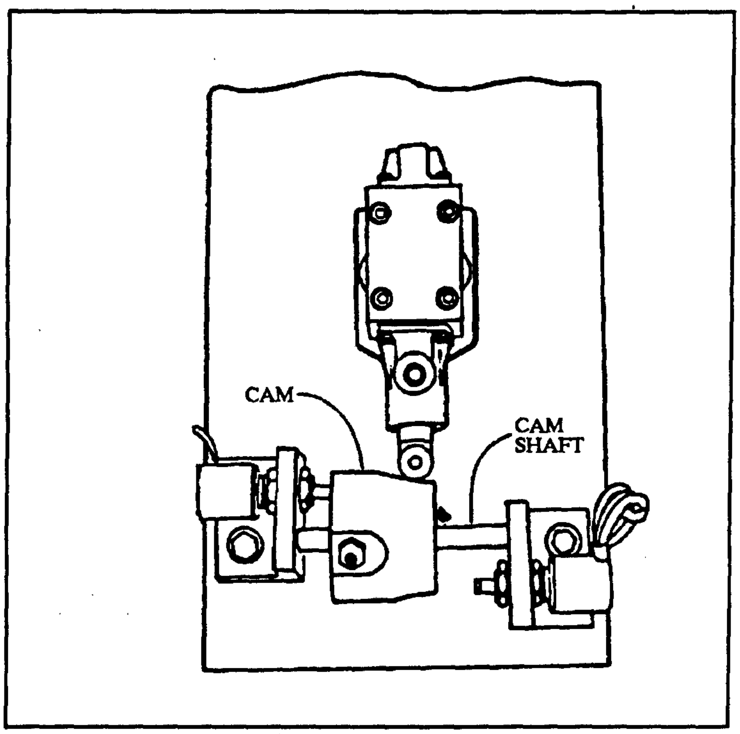

Figure 8 - Level Wind Mechanism (Partial C)

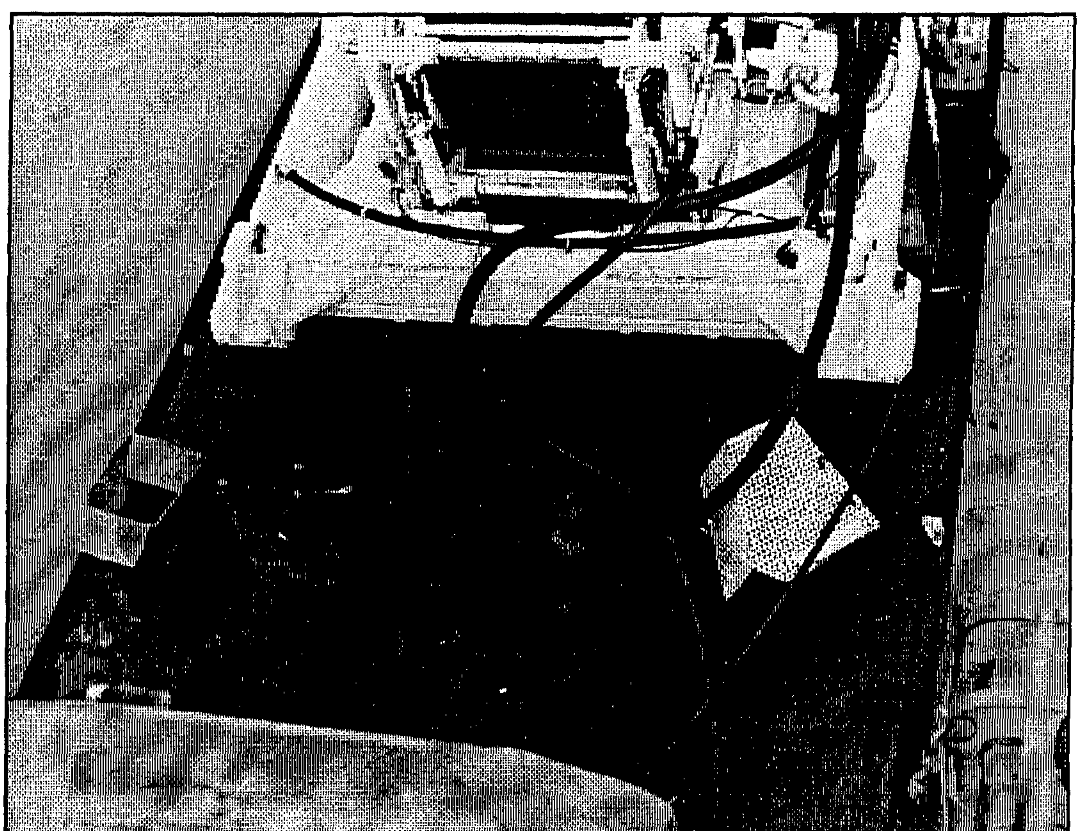

Figure 9 - Ground Hop Kit

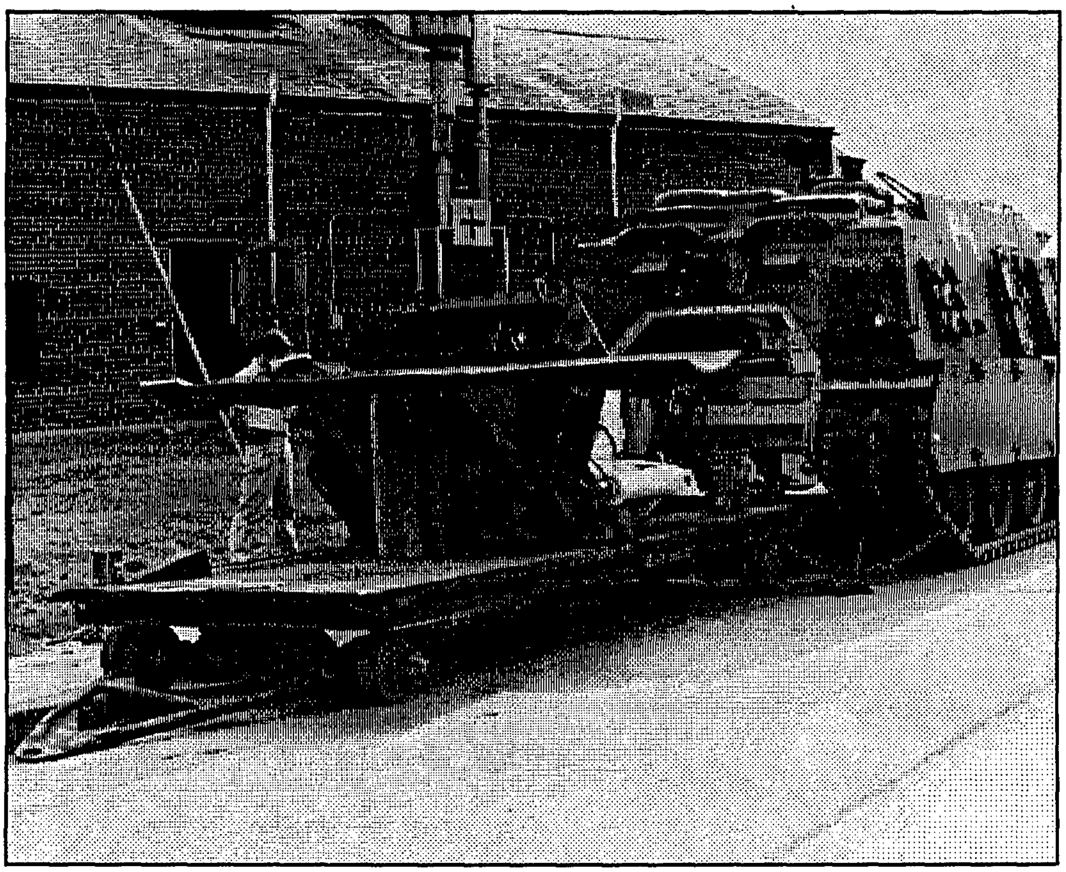

Figure 10 - Vehicle Maintenance

and check for problems. Fleet angle switches (Figure 11) prevent the angle of the cable from exceeding 20° as it is being inhauled. If angles greater than 20° are incurred, then excessive side loading to the trumpet could result in:

a. Stalling of the level wind cylinder.

b. Undue stress to level wind cam, diamond screw follower, and bracketry.

c. Excessive bending stress and abrasion to cable.

The second safety feature, the cable miswrap switches, warn the operator of possible cable *"birds-nesting."* Birds-nesting refers to underlying layers of the main winch cable acting as springs and pushing their way up to the top layer and causing a tangle. The cable basically ties itself in knots, which can cause catastrophic failures to the cable, winch housing, and surrounding support structure. To prevent this, a set of cable retention rollers are spring loaded on the back of the winch providing a clamping force against the cable. Attached to the cable retention rollers are the miswrap switches located at the rear bottom of the winch housing (Figure 12). These switches monitor the cable wrapping on the drum. If the cable rises one-quarter inch above the predetermined height, set for three levels of cable, the

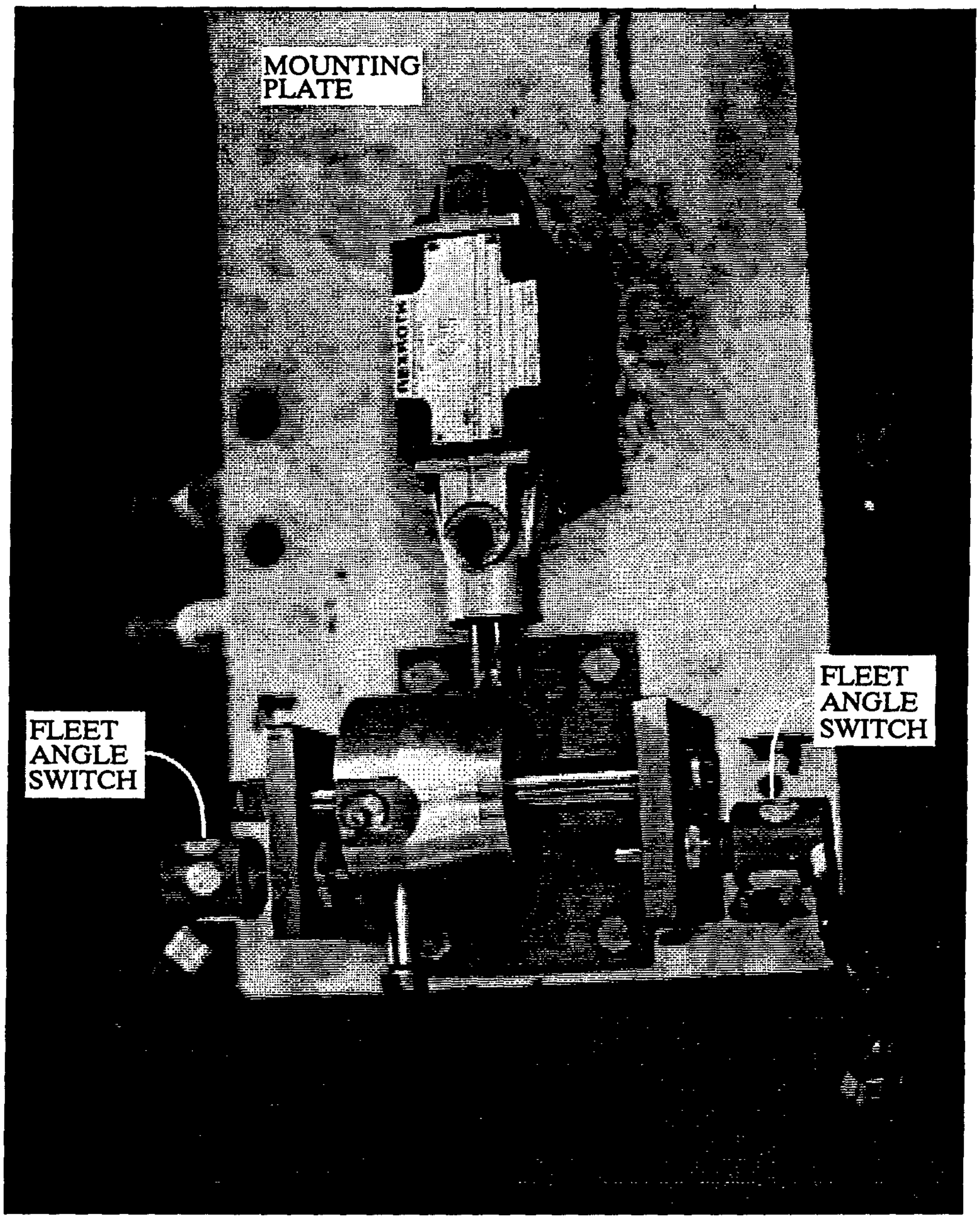

Figure 11 - Fleet Angle Switches

switches will set off the vehicle alarm. Upon testing the switches, they were found to be operating accurately. With corrections made the winch was reinstalled and P4 was ready to resume test 12 Sept 90.

5.3.2. Winch Testing

The first successful winch test was made 13 Sep 90. A series of seven pulls in the 50 to 75 percent range of full load (140,000 lbs.) was winched in over the entire main cable length of 300 feet. The oil reservoir temperature ranged from 73°F to

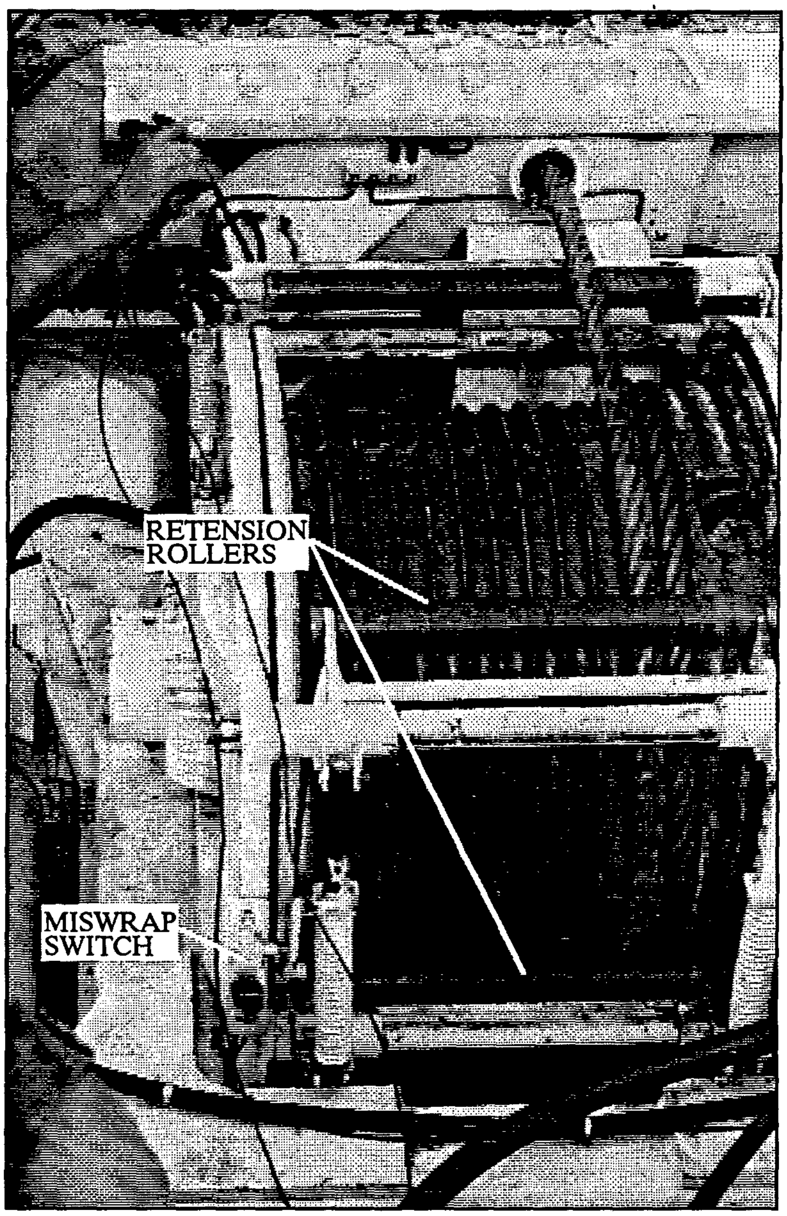

Figure 12 - Miswrap Safety Features

226°F over the five days of test. On 18 Sept 90 two pulls were made at 90% of full load. During the second 90% pull, smoke was observed coming from the nosepiece trumpet. The pull was stopped to investigate. One of the technicians revealed that the cable had started to wrap on itself but not enough to trip the miswrap safety switch. A closer inspection, again, exposed the shoulder bolt locked to the level wind cam had sheared off. The vehicle was taken back to the maintenance bay to further analyze this repeat failure. Investigation during test indicated that with no hydraulic pressure to the level wind cylinder, the nosepiece trumpet continued to move. It was assumed with the hydraulic pressure is off, the level wind cylinder should hold the winch cable in place; this did not happen. Review of the hydraulic system uncovered the absence of a cylinder lockout in the design. A 90% load equates to a 126,000-lb force on the cable and during the test the cable was being inhauled at an angle. With the load at an angle, a side load is induced (sine of the inhaul angle) forcing the level wind cylinder to either side. With the cylinder moving and the drum stationary, the bracket and shoulder bolt bridging the two together ultimately shears. The theory was validated by conducting several subtests using a portable hydraulic jack to push the trumpet back and forth with no power to the nosepiece.

Having determined the root of the shoulder bolt failure, a cylinder lockout was designed into the system (Figure 13). The figure depicts the position of the valve as installed into the system. The circled numbers correspond to the designated positions on the BMY hydraulic schematic. Reference Appendix D schematic for location. The lockout consists of a double-pilot-operated check valve (DPOCV) positioned between the level wind cylinder and the cylinder directional control valve (Figure 14). In using a DPOCV, flow out of the cylinder from either end is stopped when no hydraulic pressure is applied. The DPOCV was selected, procured, and installed. BMY was informed of the change, and they indicated that the DPOCV would be incorporated into the M88A1E1 design as drawing "BML 39594." Careful review of past level wind failures indicates that the DPOCV is a viable solution.

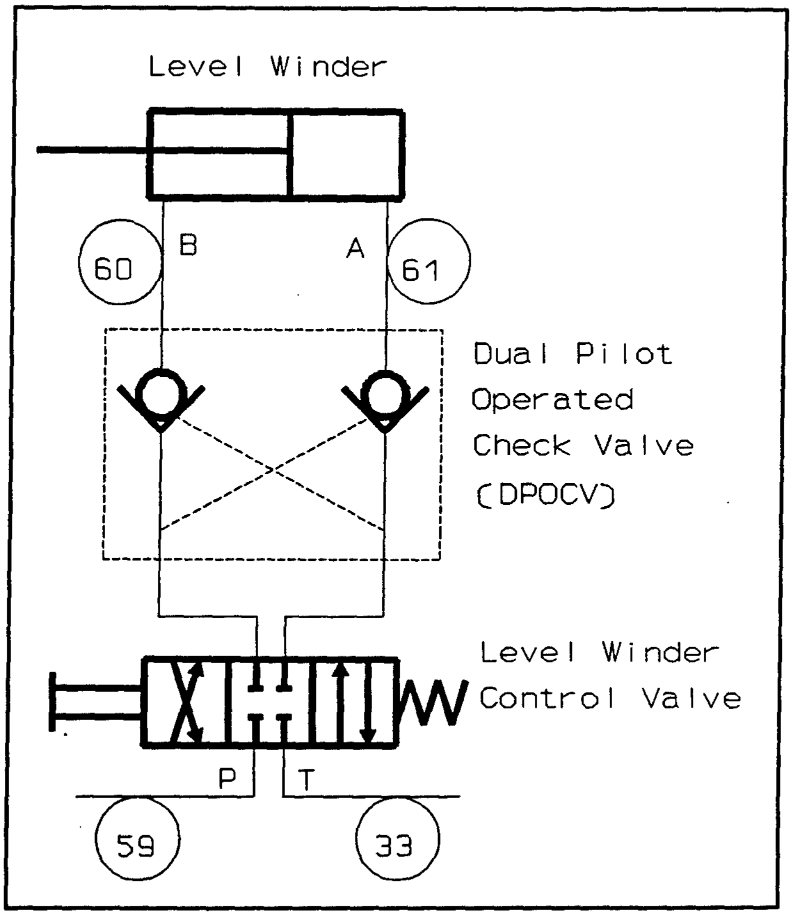

Figure 13 - DPOCV

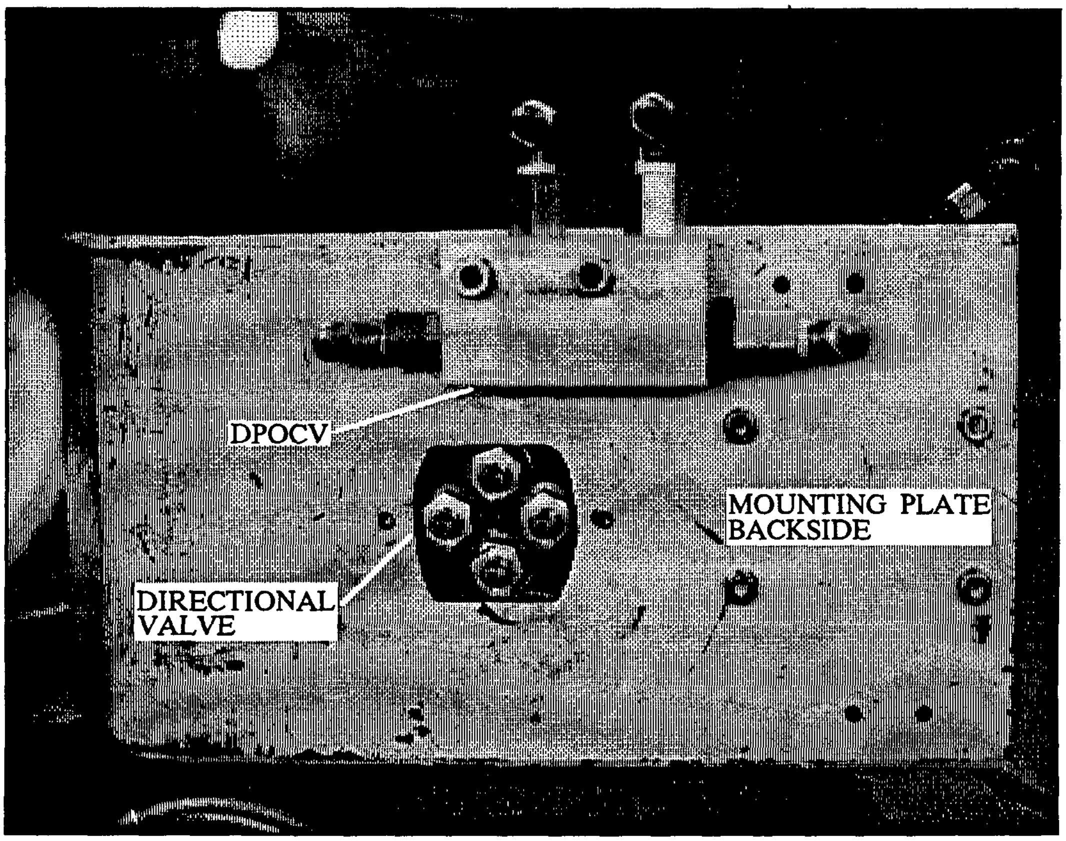

Figure 14 - DPOCV Installation

In the process of documenting the DPOCV addition to the hydraulic schematic, it was found that the schematic supplied at the beginning of test by BMY did not match the plumbing on the vehicle. With the nosepiece out, the test team scrutinized the hydraulic system against the schematic for inconsistencies. An inquiry evidenced that the schematic supplied contained changes made during the 1989 testing. At the time of the 1989 test shutdown, BMY personnel involved in the test reconverted P4 back to its original delivery design. The critical changes made by BMY involved the relocation of load sense lines and a case drain line. On 25 Sep 90, a meeting at BMY was held to discuss test issues, plumbing changes involving load sensing, and the reinstallation of hoist capabilities. BMY stated that during the 1989 testing, both hoist and main winches had performed erratically. It was thought if certain load sense plumbing changes were made, the main winch performance would be stabilized. In making these changes, the hoist function was eliminated from the system. Having determined that the hoisting function had been eliminated, BMY, T.H. Paris Co. (BMY subcontractor), and TACOM were able to produce a fix, making the hoist operational. This change involved the external mounting of a shuttle valve, relocation of a load sense line, and the addition of a drain line. The changes improved load sensing capabilities and brought the hoist back into the system. These changes (refer to Appendix D for port locations) are further explained as follows:

a. Relocation of the load sense line from the hoist and main winch DCV to the G-port of the winch motor increased load sense capability. The load was now being sensed at the point of load application and not at the remote DCV.

b. Pilot pressure line between T_1-port on the layer sense manifold and winch motor was disconnected. Two separate drain lines run back to the reservoir; one from the T_1-port, the other from the case drain running from the winch motor. The two drain lines were run back to the reservoir to reduce back pressure in the winch layer sense mechanism. This allows for better winch control. As a secondary benefit, reducing back pressure extends the life of the winch motor seals.

c. The addition of an external shuttle valve reconnected the hoist load sense to the main pump, once again making the hoist operational. The new shuttle valve has three ports. The first tees off the main winch motor G-port with a load sense line back to the shuttle valve (Line 36 on the schematic). The second shuttle valve port runs a load sense line to the hoist LS-port on the DCV. The third port, exit side of the valve, is routed directly to the load sense port on the main system pump. With this plumbing configuration, both hoist and winch loads are sensed. As the load increases, it is sensed directly back at the main pump. BMY and CSTA personnel made the plumbing and valve changes at APG. Appendix I consists of a revised hydraulic schematic with updated plumbing.

By 16 Nov 90, the hydraulic alterations were completed, and the ground hop procedure was begun on P4. With the nosepiece out, the cable was paid out one complete wrap. It was noticed that the directional valve cam follower for the level wind was riding in the extreme positions of the cam, causing the cylinder to lag behind (Figure 15). This condition indicated that more flow was being required to

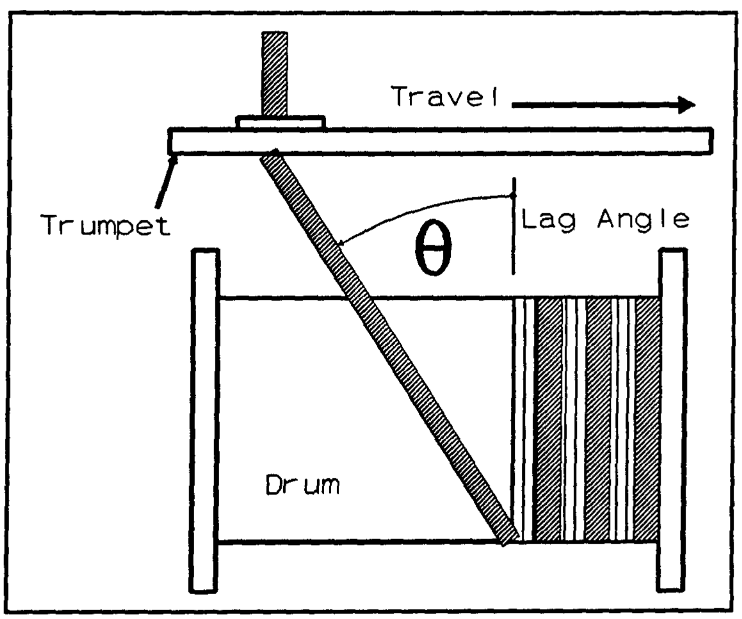

Figure 15 - Level Wind Lag

keep the level wind synchronous with the winch drum. Maximum in and out positions of the valve equates to maximum valve opening. Three separate attempts of paying the cable in and out resulted in the level wind lagging, which tripped the fleet angle switches causing the system to shut down. In order to fully open the level wind cylinder directional valve and supply more flow, the cam was moved closer to the directional valve. This change resulted in a full open position for the directional valve. The adjustment was made by loosening the slot mounted fleet angle switch brackets and pushing them forward. As a result of this change, the fleet angle switches did not trip the vehicle alarm, and the lagging problem appeared to be solved.

On 26 Nov 90, P4 was taken to the ML for the resumption of winch testing. The cable was paid out only five feet, and the warning buzzer in the vehicle went off. It was concluded that the fleet angle switch monitoring the angle of cable inhaul had been tripped. The cable was brought in a foot and then paid out. After another fifteen feet, the alarm again sounded. A visual inspection confirmed that the cable was paying straight out, and the remaining cable appeared to be properly spooled on the drum. All preceding failures to this point of test were caused by the cable being in/outhauled at angles exceeding the limits set for the fleet angle switches. With this background information available, the system override switch was depressed to restart and continue the cable outhaul. With the cable out approximately 50 ft., two loud bangs were heard. The vehicle was immediately shut down, and it was discovered that the cable had birds-nested and kinked (Figure 16). This incident caused a major failure in the main winch housing. A housing tie-rod on the back of the winch was ripped away from the housing (Figure 17). Failure diagnosis revvealed that the cable miswrap warning switches were wired in the same circuit as the level wind fleet angle switches. No indication of excessive cable angle was noted, therefore the alarm was overridden. By overriding the alarm, the cable was outhauled under no tension and birds-nested. A misdiagnosis of the vehicle alarm warning caused the housing failure. The winch could not be fixed due to extensive housing damage; consequently, the entire winch was removed and replaced with the winch from vehicle R2.

Cable birds-nesting continues to be a potential problem for the M88A1E1. Although it was an operator failure, the overriding of the warning signal is a situation that could arise in the field. Should this situation occur, the potential for damage must be minimized. The addition of breakaway bearing caps for the tie rods is a possible solution. Future winch design should consider a fail-safe design for the main winch housing and tie rods.

With the winched replaced, P4 was again ready for test starting 11 Dec 90. Six successful payin and payout winch tests were performed at 75% (96,000 lbs.) of full load. Four tests before noon that day resulted in a final reservoir temperature of 214°F. The vehicle was allowed to cool for two hours to a reservoir temperature of 167°F. The final two winch tests performed concluded with the reservoir at 204°F. On 12 Dec 90, three additional winch tests were run at 90% (126,000 lbs.) full load.

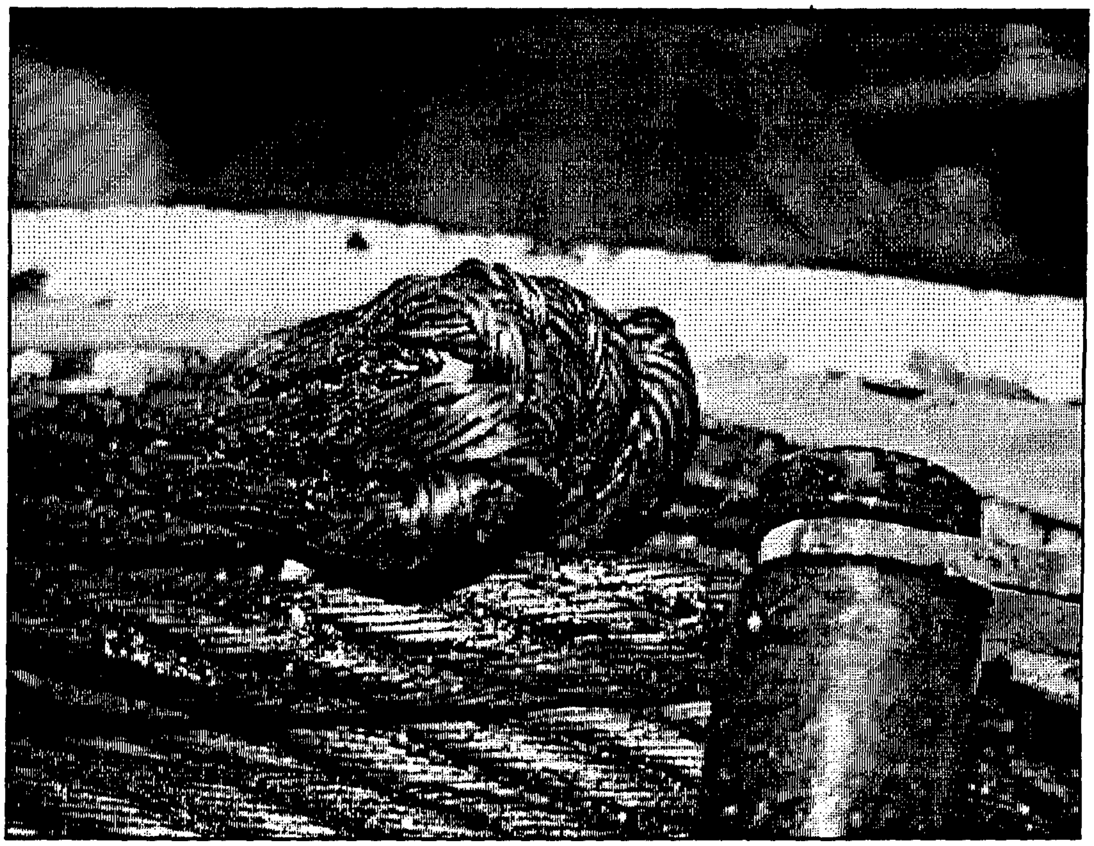

Figure 16 - Birds-Nested and Kinked Cable

The load was inhauled for 25 ft., then reduced to 65,000 lbs. for the remaining length of cable. Reservoir temperature at the start was 79°F and finished at 214°F. The vehicle cooled for two hours dropping the reservoir temperature to 177°F. When the vehicle engine was started, the charge pump recirculation dropped the reservoir temperature from 177°F to 159°F. The load was increased to 100% (140,000 lbs.), and two pulls were made finishing with 204°F in the reservoir. Two winching tests were successfully made at 110% of full load (154,000 lbs) on 13 Dec 90. The reservoir temperature started at 74°F and ended at 195° F. To this point of test, fifteen consecutive pulls were made without any problems. It should be noted that during original 1989 testing, seven consecutive pulls were the most completed at any time before the system broke down. These tests completed the winching portion of the ST. The system changes made were successful, and the winch was in/outhauling over all ranges of loading without any problems.

5.3.3. Hoist Testing

Before starting the hoist tests, the vehicle was brought to the lift and tie-down pad to determine the hoist capability, since the system was now operable because of hydraulic plumbing changes. The boom was extended on vehicle P4, and a load cell was attached to the hook block. Spreader cables were then attached to the load cell and shackled to "T" slots in the lift and tie-down pad (Figure 18). The spade

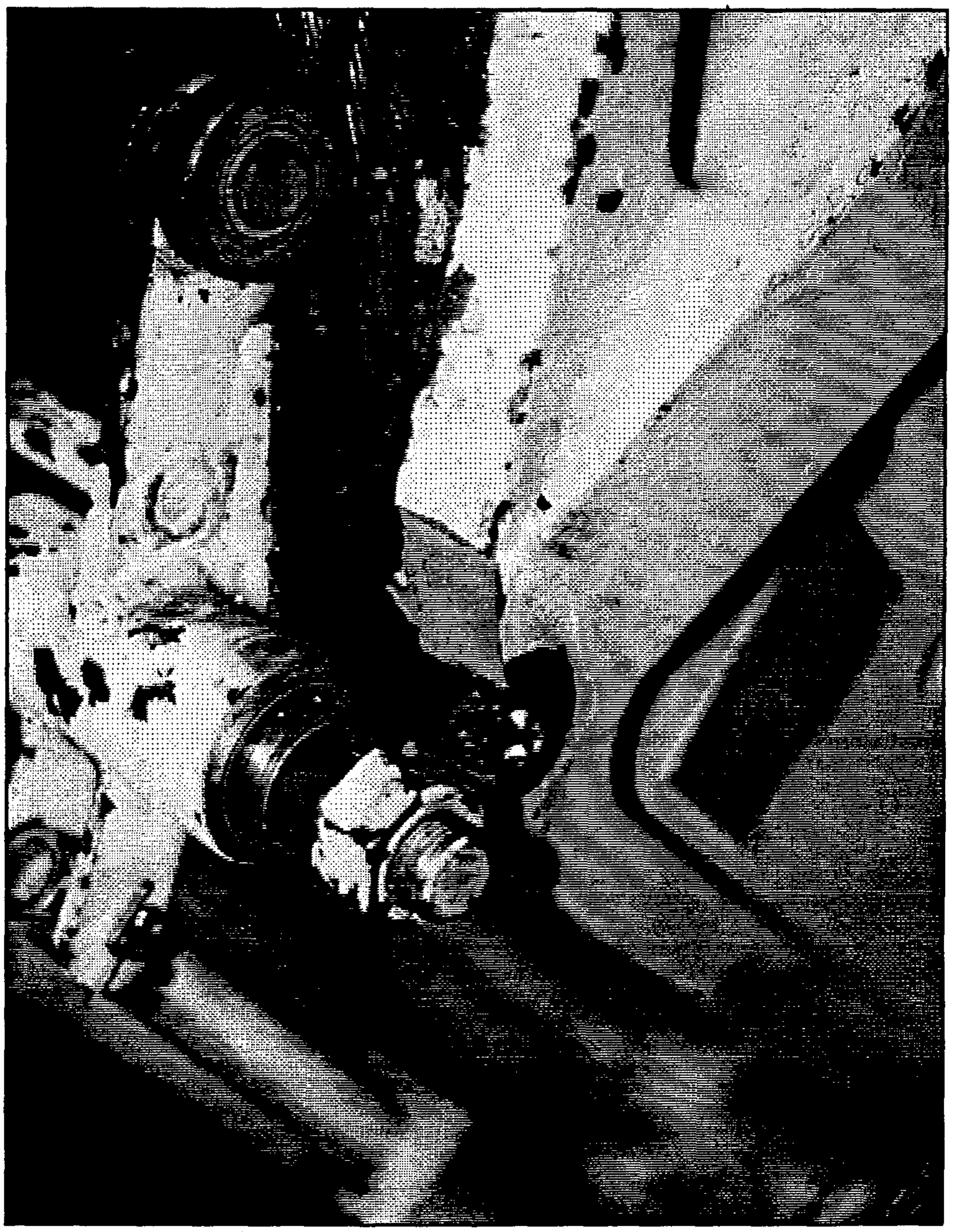

Figure 17 - Housing Fracture

was lowered to provide a mechanical advantage and vehicle support. The hoist was actuated, and a load of 87,000 lbs. was observed. The load was backed off as to not damage the boom, since the force observed was greater than the 70,000 $\pm$ 10% requirement. The hoist winch motor is a fixed volume unit that increases torque through increased pressure; therefore, to limit the hoisting capability, incoming pressure must be reduced. However, if the pressure were to be set for 77,000 lbs. with the hook at the top to meet the requirement, a higher lifting capacity would be realized with the hook all the way out. This greater load may exceed the vehicle's boom specification. BMY was contacted on this matter. They indicated that this

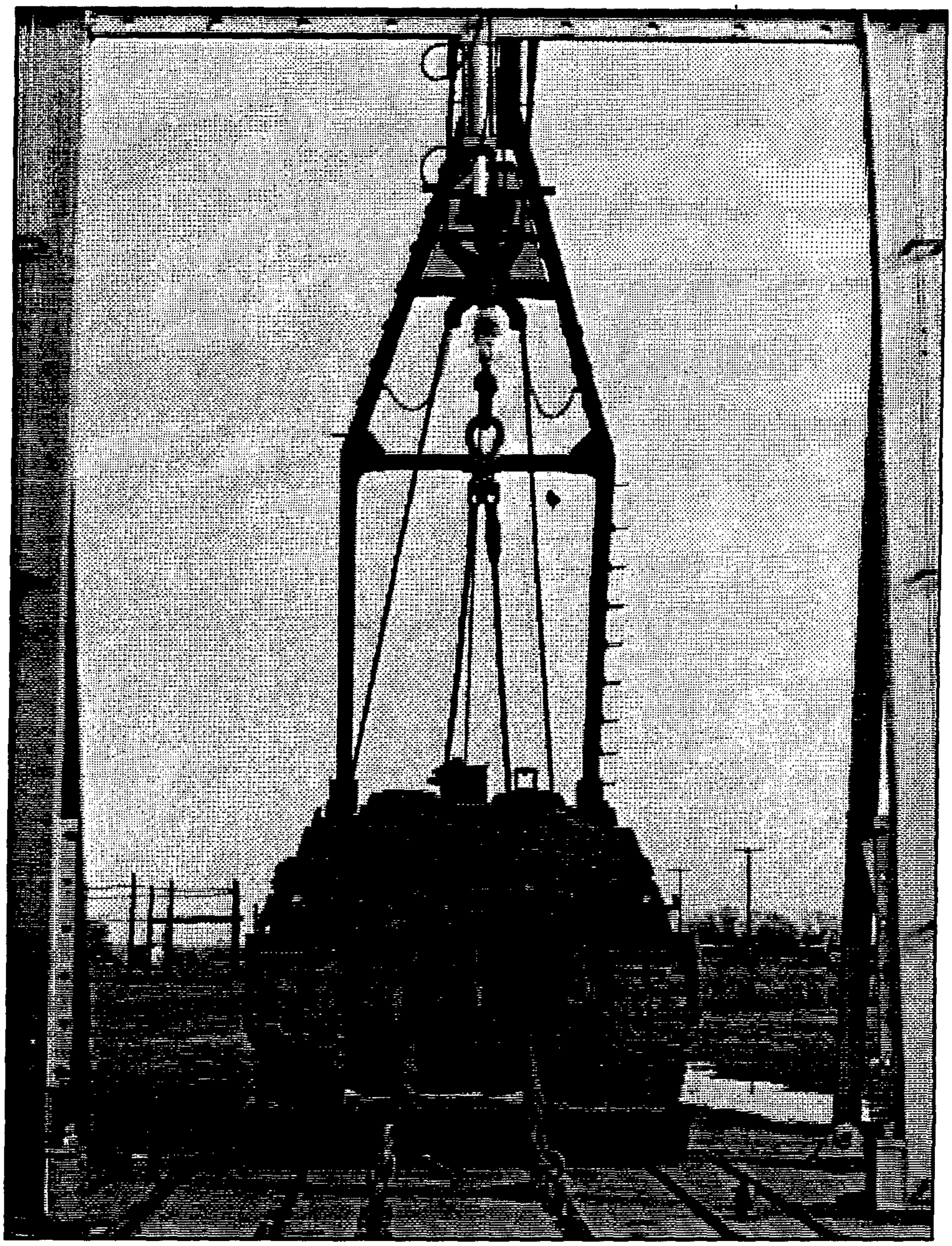

Figure 18 - Hoist Test Setup

redesigned the boom and now has sufficient strength to handle the increased lifting capability.

Once the DCV was set at the proper reduced pressure value, baseline hoisting was started. No overheating of the hydraulic fluid was encountered in the ten lift/lower trials. Problems, however, were noticed with the hook block, as it had to be replaced twice because of excessive wear. The original BMY hook block design (Figure 19) for the M88A1E1 incorporates a bushing for the sheave shaft bearing

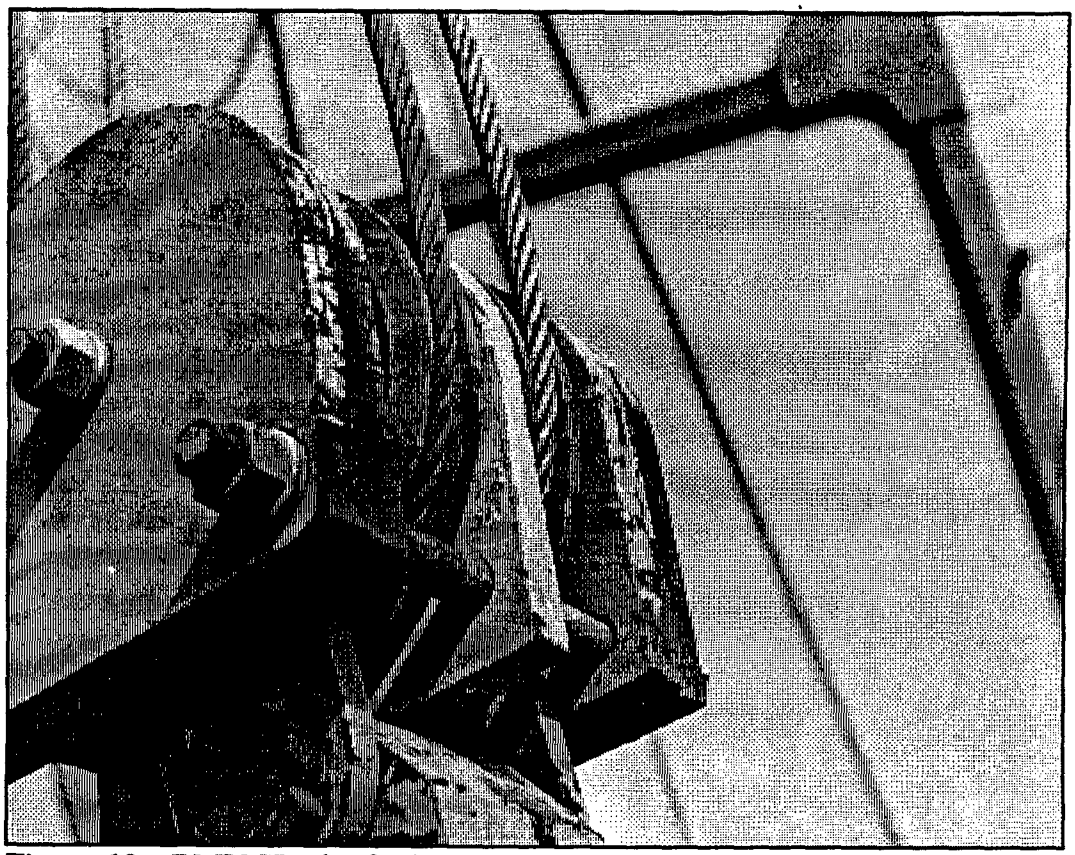

Figure 19 - BMY Hook Block

surface. The block was not designed for continuous use and consequently encountered failures during test. Continuous use under full load caused extreme heat buildup and wear to the rotating components. BMY has discussed a new

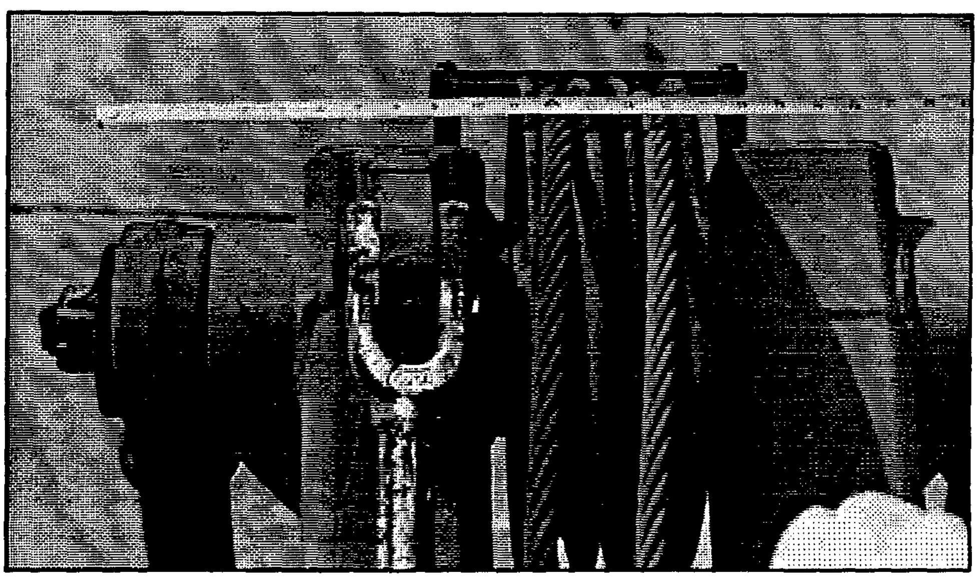

Figure 20 - Boom Sheave and Deadman

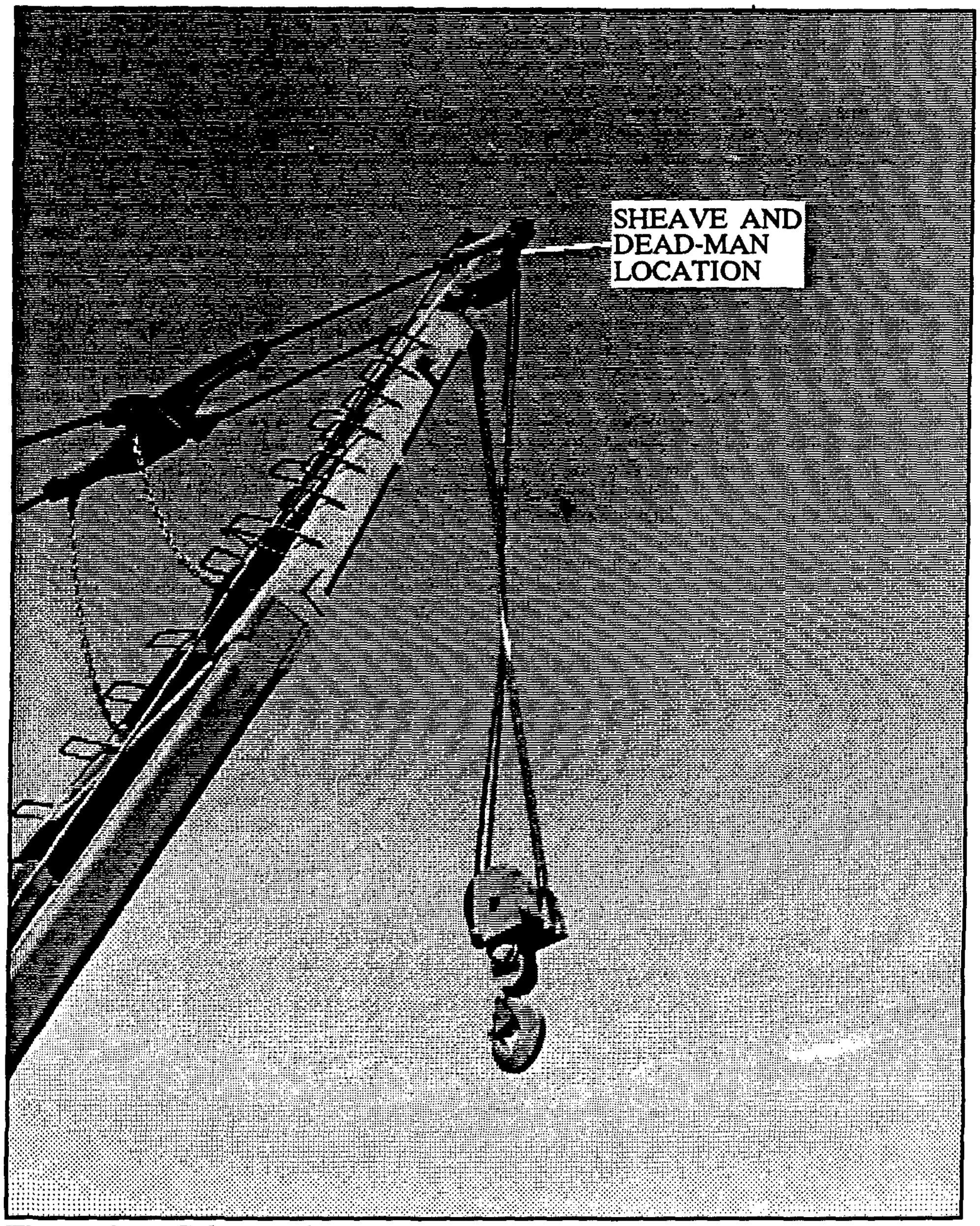

Figure 21 - Cable Twist

design with the Johnson Block Company, in which roller/ball bearings replace the bushing.

The current hoist winch cable deadman location and boom sheave design has also proven to be inadequate (Figure 20). The deadman as positioned on the boom causes the hook block to twist during hoist operation (Figure 21). Twisting of the block causes the boom sheave to bind and cable to wear. In an attempt to minimize twisting, the deadman was rewelded in a new location (Figure 22). Furthermore, binding and cable wear are realized, due to the unequal spacing between boom and hook block sheaves. The angle created during operation induces side loading on the

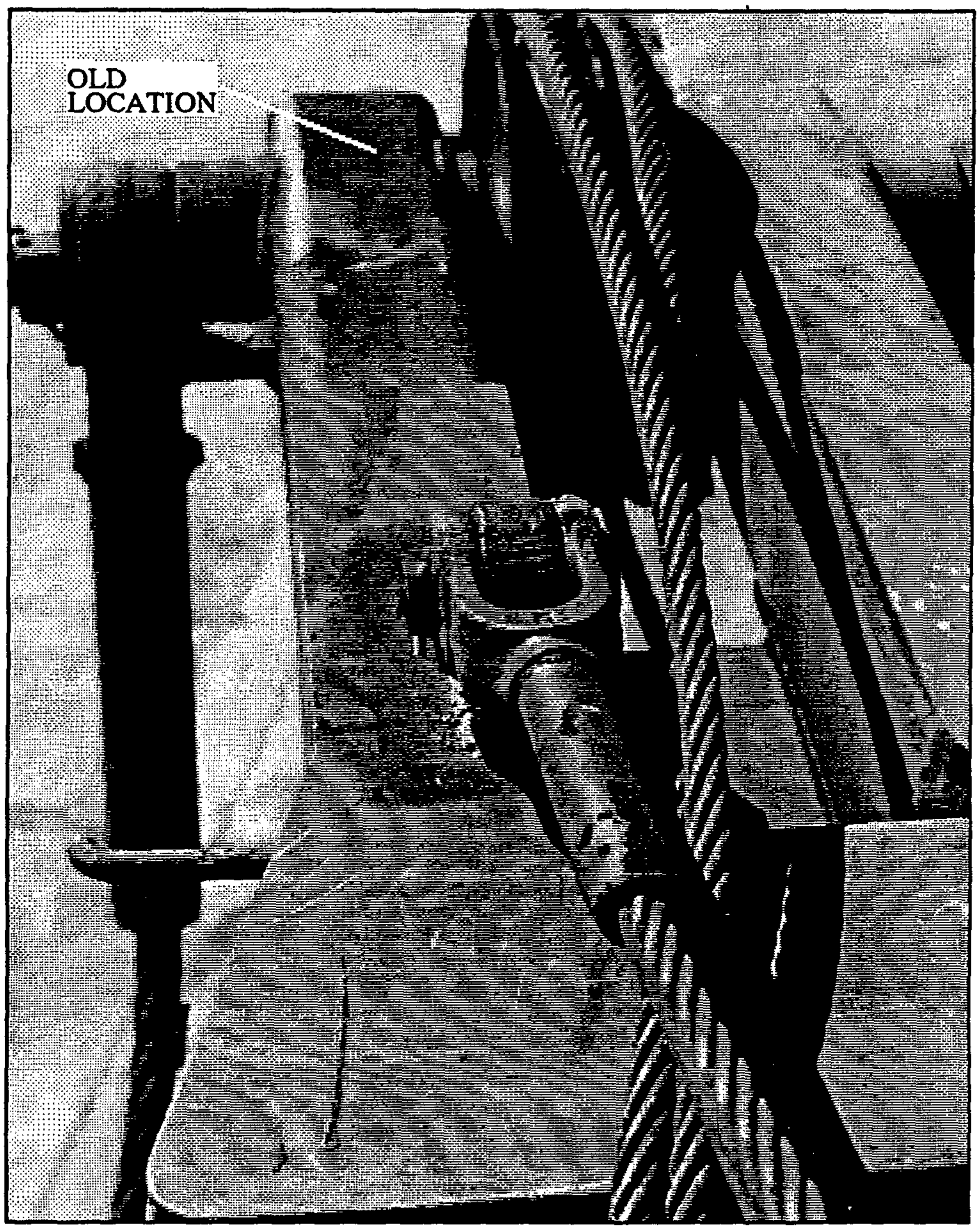

Figure 22 - Relocated Dead-man

sheaves. The most noticeable problem occurs during a 70,000 lb. lift. If the hoist was stopped at an intermediate point of travel and restarted, the load would not move due to the frictional buildup.

5.3.4. Auxiliary Winch Test

Aux winch tests were conducted concurrent with duty cycle tests, and no failures were encountered; however, an in-depth review of the system design has concluded that the aux winching cycle is a source of significant heat generation. The main and aux winches are connected in the same circuit and system pressure is supplied to both winches. The problem occurs when the aux winch is pulling out the main winch cable. The main winch only requires 500 psi to spool during this function, while the aux demands 4,000 psi. Needing only 500 psi, the main winch pressure relief dumps the remaining 3,500 psi directly to heat which is added to the hydraulic system.

5.3.5. <u>Duty Cycle Testing</u>

The duty cycle test had a twofold purpose:

a. The duty cycle, as defined in the PD, had never been tested during the previous TFT and PPT. The ST would prove out the adequacy (too stringent or too simple) of the duty cycle.

b. The data obtained from the duty cycle test would be the determining factor as to the need for a hydraulic oil cooler.

As defined in the PD, the duty cycle would entail three full hoists at 70,000 lbs. (Figure 23), and one full winch at 140,000 lbs. (Figure 24) over the first 25 feet of cable and the remainder of the cable length at 69,000 lbs., to be completed in one hour. The aux winch would be used to deploy the main cable. This cycle would be run for four consecutive hours.

Duty cycle tests began 6 Mar 91. At the end of first hour, the reservoir temperature read 155°F. The second hour of the cycle was completed, with the temperature rising to 203°F. At the start of the third hour, the BMY hook block failed on the first hoist, due to side-loading wear and heat buildup. This was the last workable BMY hook block available. All five prototype hook blocks had failed at various times during the ST. BMY did not have any extra blocks or spare parts available for support. In order to continue testing an off-the-shelf commercial hook block was procured in Baltimore, MD. The unit purchased was a Johnson block (Figure 25) rated at 30 tons with a 4.0 safety factor, which equates to a loading capability of 120 tons. The BMY hook block was rated at 35 tons with a 3.5 safety factor for a 122.5 ton capability. The Johnson distributor guaranteed the 30-ton block for a 35-ton application.

With the new block installed, a second day of duty cycle tests were run. The reservoir temperature started at 84°F and with only three hours of test completed, the reservoir temperature had skyrocketed to 243°F. After consulting with the motor and pump manufacturer, (Rexroth Corporation) it was decided that testing would be halted when temperatures rose to 250°F. Temperatures over 250°F would damage pump and motor seals and accelerate component wear due to fluid breakdown. The third day into the duty cycle resulted in two more hours of testing before having to shutdown because of overheating. It became apparent during this testing that a full duty cycle could not be completed while maintaining the reservoir temperature at 170°F. After three hours of testing, reservoir temperatures were reaching 250°F on days when they began at 40°F. The vehicle was able to handle the duty cycle without fail, with the exception of the hydraulic fluid overheating.

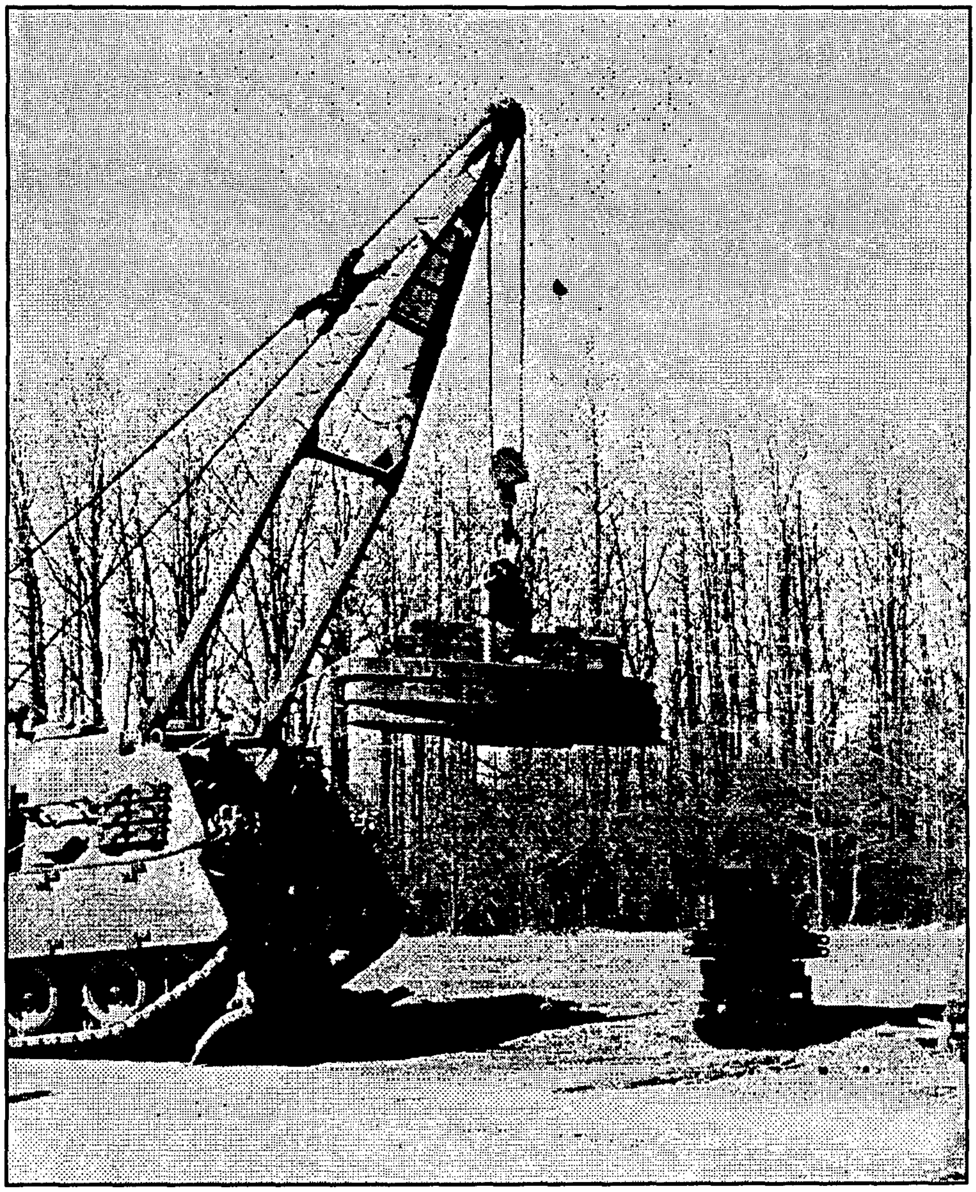

Figure 23 - Hoist Test

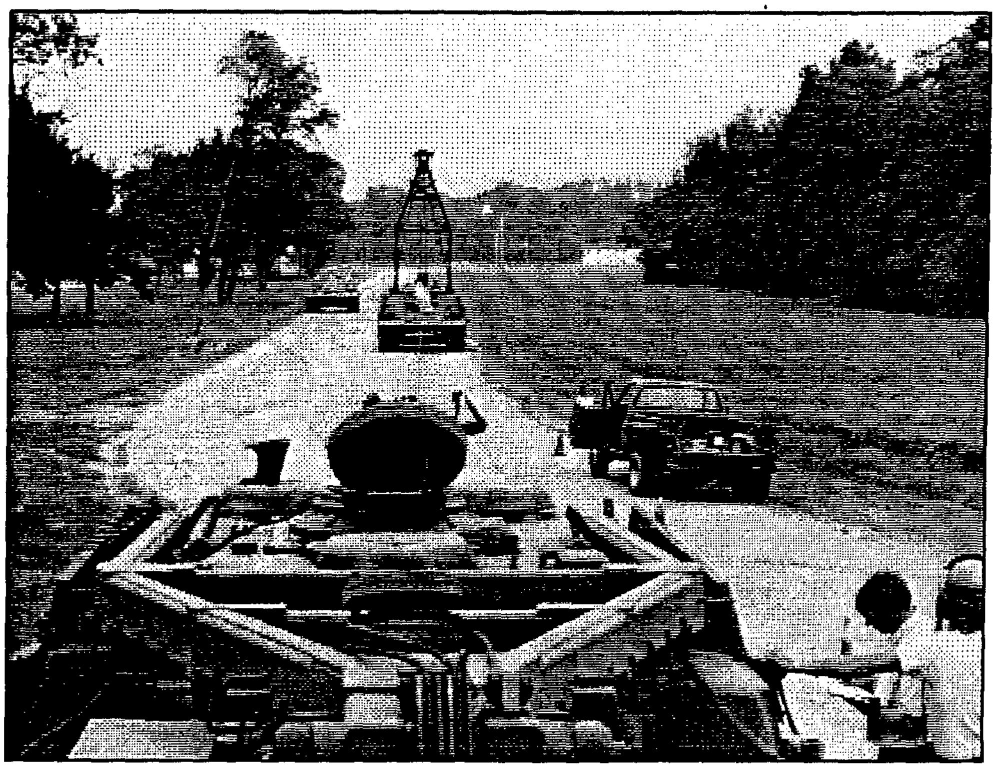

Figure 24 - Winch Test

5.4. Test Analysis

The main problem plaguing the M88A1E1 is the excessive heat buildup in the reservoir. All other problems were satisfactorily repaired to complete the ST. Appendix J encompasses a complete listing of parts added to P4 throughout the ST. With respect to the heat problem, several meetings were held between BMY, TH Paris Co., and TACOM, to determine what system modifications were needed to maintain reservoir temperatures at 170°F. Two possible solutions recommend were:

a. Three-pump system

b. Hydraulic oil cooler

5.4.1. Three-Pump Design

Section 5.3.4. detailed the operation of the aux winch and explained the excess generation of heat at the main winch. The design was reviewed, and it was concluded that if the aux winch could be isolated in the hydraulic system (i.e., the main and aux winch having separate pumps), the problem would be eliminated. The excess 3500 psi will not be seen and therefore not rejected in the form of heat, thus eliminating a system inefficiency. The T.H. Paris Corp. was the supplier of the original Rexroth pumps for the M88A1E1 and were brought in as a consultants for

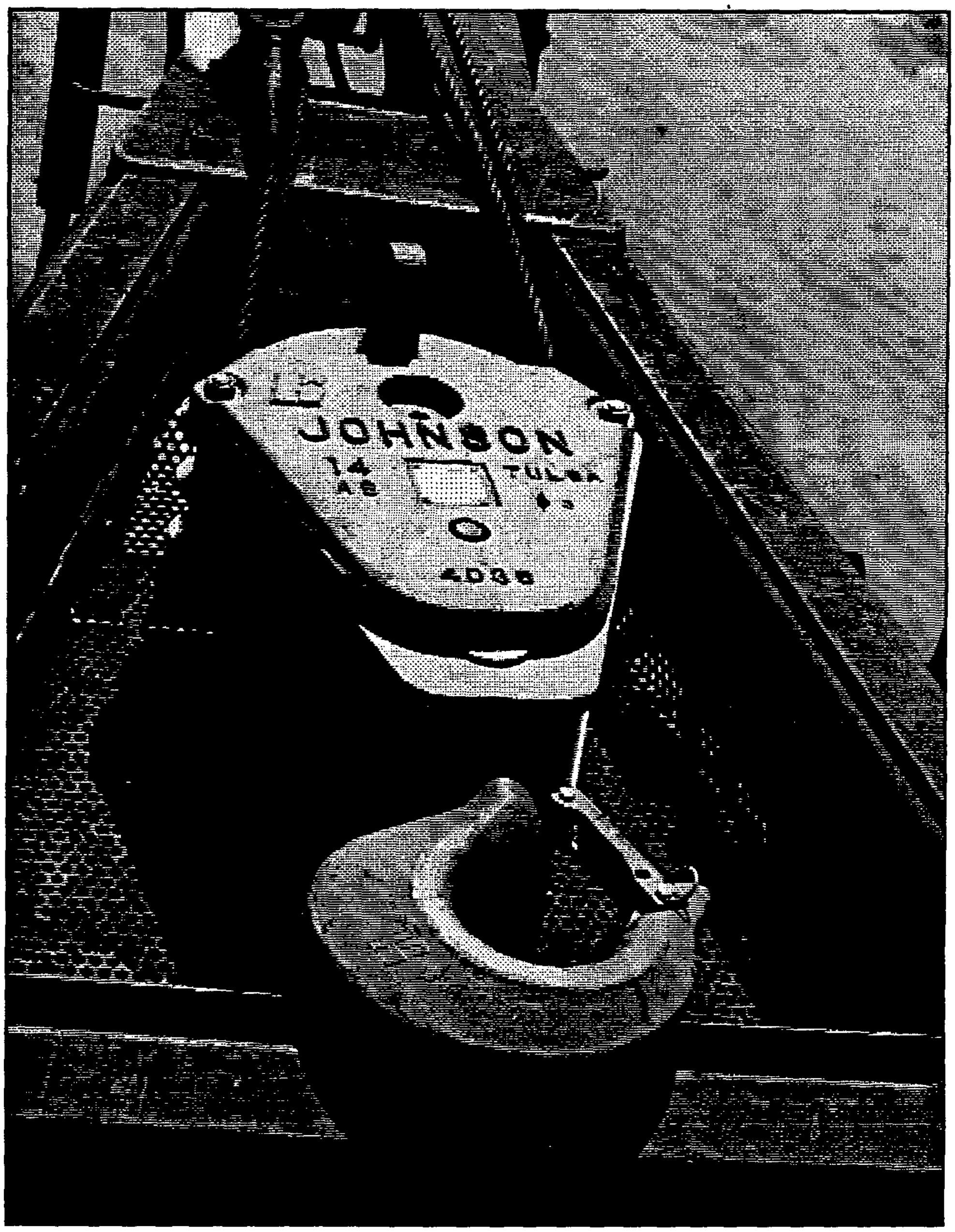

Figure 25 - New Hook Block

the installation of the third pump. A three-pump design was developed and submitted to BMY for their review and approval. With all parties in agreement, the three-pump system was ordered and installed (Figure 26). Drawings and parts lists detailing the three-pump installation are included in Appendix K.

Upon completion of the installation, three days of winch tests were conducted. In comparing the results of the two-versus three-pump system for heat generation, the three-pump system reduced heat gain during the aux winch cycle by approximately 35%. The 35% savings allowed the test to run into the fourth hour of the duty cycle. Although achieving the savings, the final reservoir temperatures were still attaining 240-250°F.

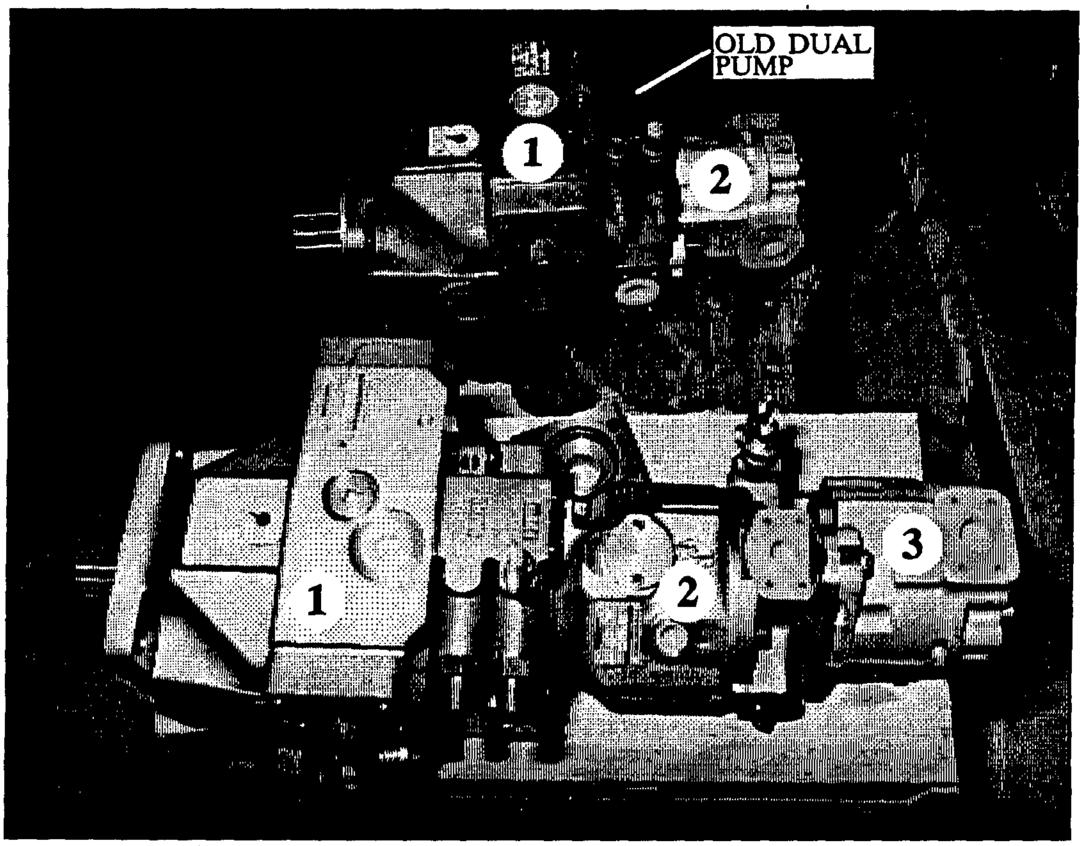

Figure 26 - Three Pump Assembly

5.4.2. Hydraulic Oil Cooler

Based on testing in 1989 and the ST, it has been concluded the M88A1E1 cannot perform continuous hydraulic functions without hydraulic oil temperatures exceeding the maximum allowable. The PD requires the completion of a full duty cycle while maintaining an oil reservoir temperature of 170°F. As a consequence of the test:

a. Only three hours of the four-hour duty cycle could be completed (two-pump system).

b. Oil reservoir temperatures rose as high as 250°F after three hours of test, far exceeding the 170°F limit.

The only possible solution to complete a duty cycle and maintain a 170°F reservoir temperature is to install an oil cooler. System optimization for piping, valves, and the installation of a three-pump unit will control temperatures for a short time; however, over continuous operation, temperatures will continue to rise until maximums are exceeded. An analysis of the data taken during ST has determined that under maximum heat gain conditions (continuous winch operations at 120°F ambient air temperature), the vehicle will require a cooler sized to reject a minimum of 660 BTUs/min. It should be noted that the ST was performed in temperatures

50-60°F not 120°F temperatures. Appendix L consists of a thermal analysis, sizing data, and an example cooler selection for the M88A1E1 hydraulic system. Figure 27 exhibits a generic fin tube, air to oil, heat exchanger.

Adding an oil cooler to the M88A1E1 will require vehicle modifications. These modifications include:

a. Ballistic housing for the cooler.

b. Air filtration system for incoming cooler air.

c. Piping and valving interface connections between cooler and hydraulic system.

Upon successfully installing the cooler, a series of tests should be run to optimize the cooler selection. Downsizing of the cooler will be a benefit to vehicle packaging.

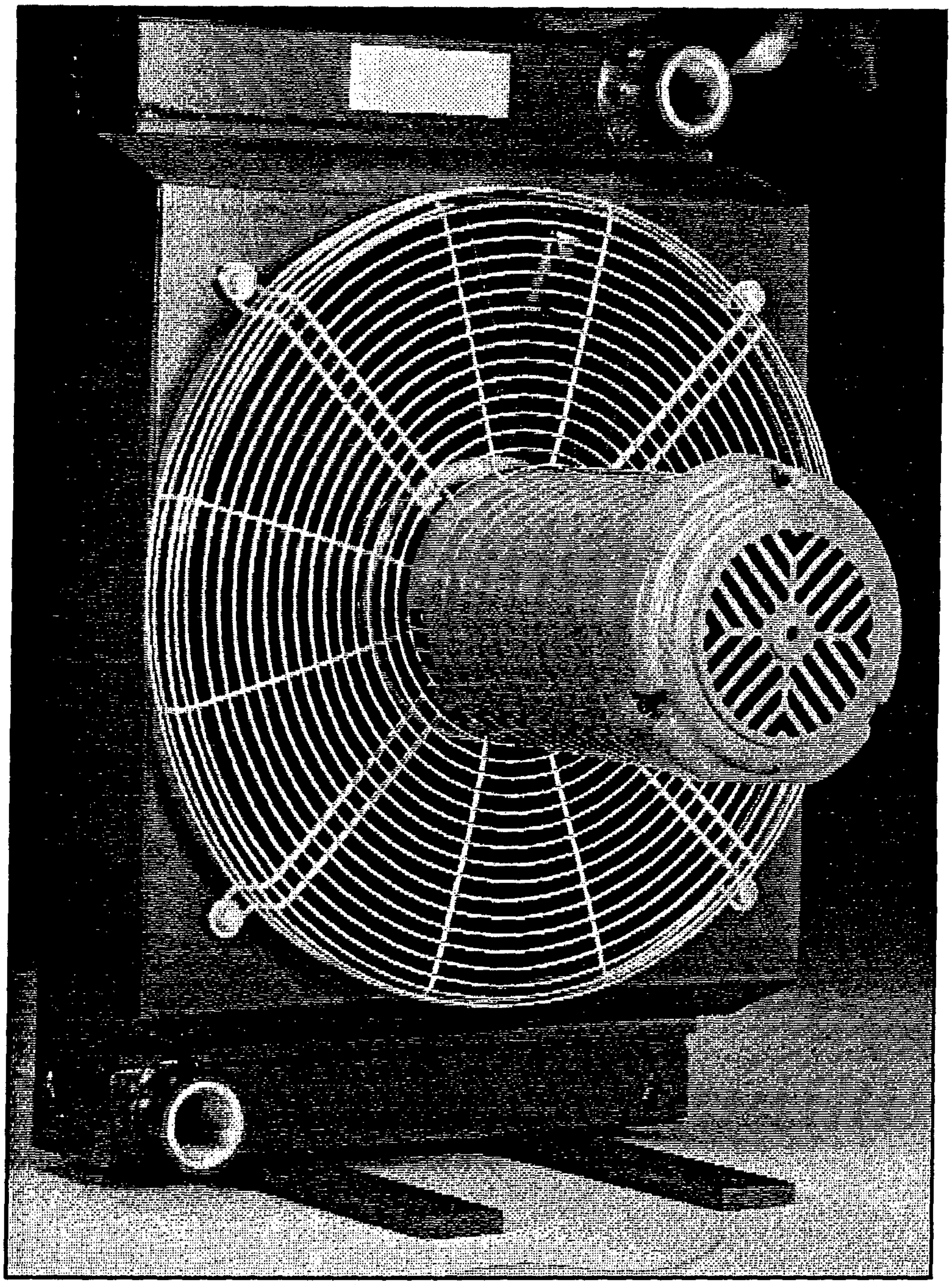

Figure 27 - Typical Air-Oil Heat Exchanger

APPENDIX A

Program Personnel

M88A1E1 IMPROVED RECOVERY VEHICLE
SUMMER TEST

PROGRAM PERSONNEL

NAME	AFFILIATION	PHONE
Gerry Yursis	CSTA	301-278-5612
Jerry Workinger	CSTA	301-278-7722
Jeffrey Greenberg	CSTA	301-278-7734
William Fraser	CSTA	301-278-7720
Paul Gross	CSTA	301-278-3801
John Roberts	TACOM	313-574-6438
Ronald Chapp	TACOM	313-574-5061
Steven Knott	TACOM	313-574-5061
Robert Robinson	TACOM	313-574-6468
Daniel Lam	TACOM	313-574-6468
Leslie O'Neal	TACOM	313-574-6172

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George See	BMY	717-225-3400
Chuck Lewis	T.H.Paris	717-244-0296
Robert McCutcheon	Johnson Block	918-832-8933
Thomas Lembcke	Thermal Transfer	414-554-8330
Richard Bradach	P&H	414-764-8518
Hugh Parkhurst	PAL Filters	516-671-4000
John Eleftherakis	FTI	465-744-7375

APPENDIX B

Major Test Incident Reports

TEST INCIDENT REPORT (TIR)
TECHNICAL FEASIBILITY TEST (TFT) FY89

CLASS: MAJOR

TIR#	FACAR	SUBJECT	RESOLUTION
K2-B00066	Y	Hydraulic impact wrench has low torque output	New dual stage relief valve & impact wrench proposed - no impact on summer test
K2-B00133	Y	Charge and return filters clogged	New filters installed - NFAR
K2-B00141	Y	Erratic main winch stalls	Hydraulic system relief pressure reset - NFAR
K2-B00142	Y	Faulty replacement part	Faulty cable replaced - NFAR
K2-B00143	N	Faulty replacement part	320 foot cable installed 330 feet required, accepted as is - NFAR
K2-B00145	Y	Class III hydraulic leak	New tube assembly (P/N 344N76) installed - NFAR
K2-B00155	Y	Main winch lever assembly bent	New lever installed - NFAR
K2-00156	Y	Faulty replacement part	Cable replaced - NFAR
K2-B00158	Y	Class III hydraulic leak	Aux winch return line "quick disconnect" replaced - NFAR

K2-B00147-67	Y (AR)	Eleven TIRs detail hydraulic oil overheat problem	Overheating problem documented; no resolution at time; installation of oil cooler will resolve problem. Summer test substantiated overheat problem
K2-B00172	Y	Main winch cable pulled out from wedge	Minimum of three wraps are required for bare drum; replaced wedge - NFAR
K2-B00196	Y (AR)	Clutch housing lacks drainage capability	BMY interoffice memo dated 6 Sept 88 indicates production clutch modifications will "relocate drain plug to left of clutch; will include manual valve to mate with clutch drain circuit leading to mech compartment hull drain"
K2-B00217	Y (AR)	Electromagnetic clutch housing lacks drain plug	(Same resolution as K2-B00196)
K2-B00226	N	Hoist would not lift 35 tons	Pressure relief valve readjusted - NFAR
K2-B00427	N	Vehicles #1 & #2 plumbed differently	APU relief valves mounted at the driver's station - problem resolved - NFAR
K2-B00450	N	Faulty QD	QD replaced - NFAR
K2-B00630	N	Adjust boom relief valve	Boom relief valve adjusted - NFAR
K2-B00650	N	Power takeoff (PTO) would not engage	Replaced PTO splined shaft - NFAR

K2-B00678	N	Noise coming from PTO assembly	New PTO clutch required, PTO replaced - NFAR
K2-B00778	N	PTO clutch would not engage	PTO replaced - same as K2-B00678
K2-C000008	N	Aux cable wrap problem	New modified aux winch installed - NFAR
K2-C000010	N	Class III leak - main winch	New cap plug installed - NFAR
K2-C000011	N	Main winch cable damaged	Replaced cable - NFAR
K2-C000016	N	Level wind damaged	Sheared mounting bolt replaced - level wind failure problem solved during summer test
K2-C000038	N	Faulty filter valve manifold	Faulty filter valve manifold replaced - NFAR
K2-C000040	N	Erratic main winch behavior	Load sense line relocated during summer test, erratic behavior eliminated - NFAR
K2-C000041	N (AR)	Aux boom inadequate	Not addressed during summer test - BMY to address under contract two modifications
K2-C000042	N	Hoist sheave seized up	New hoist sheave design to be installed by BMY under contract two

K2-C001030	N (AR)	Hydraulic boom inoperative	Cold operation of hoist indicates hoist will not work. BMY to address problem under contract two refurbishment. Oil preheat is required.

Y = Yes AR = Action Required

N = No NFAR = No Further Action Required

APPENDIX C

1990 Summer Test Directive

M88A1E1 HYDRAULICS TEST DIRECTIVE

1. Authority: This test directive provides authority to the Aberdeen Proving Grounds (APG) to perform the testing contained herein on the M88A1E1 hydraulics systems.

2. Test Responsibility: M88A1E1 testing shall be conducted under the direction of the Systems Planning Branch (AMSTA-ZDS), TACOM, Autovon 786-5061. Any requests or directions received by the test site from any other office or agency shall not be honored unless specifically authorized by AMSTA-ZDS.

3. M88A1E1 Test Vehicle: Presently five (5) M88A1E1 prototypes are available at APG for testing. Vehicle #4 shall be used for the hydraulic tests. Vehicle #5 shall be available as needed.

4. Purpose of Test: As a result of PreProduction Tests (PPT), it has been determined that the M88A1E1 has experienced hydraulic overheat problems. Also an excessive buildup of pressure in system pressure relief valves has caused malfunctions. Both problems appear to be the source of related hydraulic problems. The purpose of this test is to:

a. Identify current M88 hydraulic problems.

b. Determine the source of the problems.

c. Recommend possible corrective actions.

5. Governing Documents: All testing shall be conducted in accordance with this test directive.

6. Reports and Correspondence: All reports and correspondence issued as a result of this directive and testing shall include, as reference, test title, TACOM Project Number, and name of system tested.

7. Test Completion: The test should be completed by 31 Aug 90. All tests shall be scheduled to assure completion within the time requirement. The final test report shall be submitted within 30 working days of test completion.

8. Test Program Review:

a. An initial meeting was held between APG and TACOM, 16-17 Apr 90, to discuss the results of the PPT and to try and determine what M88A1E1 operational problems still exist.

b. TACOM has reviewed Test Incidence Reports (TIRs) to determine what actual problems were reported during PPT. Based on the TIR review and conversations held at APG, basic hydraulic problems have been identified. TACOM has written this test plan to determine if these problems still exist, and if they do, what steps might be taken to correct the problems.

c. If deemed necessary by the sponsoring activity (TACOM, AMSTA-Z-IRV), a test review meeting shall be called at a location to be determined. The conference may include, but is not limited to the discussion of the following items:

(1) Test delays and their causes.

(2) Deviations from the test directive.

(3) Discrepancies discovered during testing that APG or TACOM determine to be of major concern.

(4) Miscellaneous items of significance.

9. Spare Part Support: CSTA shall supply all necessary spare parts as available to keep tests running on schedule. Spare parts shall include but not be limited to hydraulic oil, quick disconnect fittings, pressure hose, valving and actuators.

10. Test Procedure:

a. System Verification: The system developer, BMY, shall supply system installation drawings and/or available manuals for all hydraulic circuits. A TACOM hydraulics technician shall use the installation drawings to verify the actual hydraulic circuitry plumbed into vehicle #4. Any discrepancies shall be marked on the drawings. A TACOM hydraulics technician shall be on site to assist in the system verification prior to testing.

b. System Inspection: Each hydraulic system shall be further inspected to insure its' integrity. All hoses shall be connected, systems shall be full of oil, and all moving parts (motors, pumps, actuators, etc.) shall be operable.

c. Test Reporting: All serious test incidents shall be immediately reported to AMSTA-ZDS by telephone (AV 1-786-5061). CSTA shall submit raw test data directly to TACOM, AMSTA-ZDS. A cover letter shall be attached briefly describing the data being submitted. Data shall be issued in accordance with distribution instructions contained in paragraph 13.

d. Performance Phase:

(1) A series of operational tests shall be performed to:

(a) Verify if defects found during PPT have been corrected or still exist.

(b) Determine the cause of current hydraulic problems.

(c) Determine if other major problems exist that may not have been uncovered during the PPT.

(2) The following hydraulic circuits shall be identified and tested:

(a) pump/filter main charging loop

(b) hoist/winch

(c) main/auxiliary winch

(d) boom

(e) spade

(f) stayline

(3) Parameters to be monitored during circuit testing are:

(a) reservoir temperature

(b) flow rate immediately after the main pump

(c) pressure and temperature exiting main pump

(d) for each circuit containing actuators, pressure and temperature exiting one (1) of the actuators.

(e) for each circuit containing a pressure relief valve, pressure and temperature immediately downstream of the valve.

(f) pressure immediately upstream of the valve, to allow for pressure drop determination.

(4) The raw test information gathered shall conform to the format given in exhibit #1. The top third of the page shall contain a sketch of the circuit being tested. The sketch shall describe all elements of the circuit to include; pumps, motor, actuators, valving, reservoir, test gauge locations, and hose sizing. The lower two thirds of the page shall list the test results in the format of exhibit #1.

(5) Vehicle #4 shall be tested at conditions that produce maximum work or heat for the purpose of verifying the operational capabilities of the hydraulic system. The following operational conditions shall be tested:

(a) Engine Speed: For each series of tests, the engine shall be tested at tactical idle or 1800 RPMs. An idle or full RPM condition shall have a minimum duration of 30 minutes. During this 30 minute interval, temperatures, pressures, and flows shall be monitored. If there are no appreciable changes in values being recorded, $\pm 5\%$, the test data may be considered acceptable. If pressure values increase more than 5% over full rpm conditions, the test shall be continually monitored. If the pressure increases 10% or more, the test shall be halted. With the use of 15 W 40 wt. hydraulic oil, the maximum allowable system temperature shall be 170 degrees Fahrenheit. Tests shall be halted above this temperature.

(b) Boom Operation: The boom shall be tested under no and full load conditions. Full load shall be 35 tons in accordance with vehicle specifications. In the raising and lowering of the boom, the boom shall be intermittently stopped and started. It is necessary to check the ability of the boom to be restarted from an intermediate point of travel.

(c) Main/Aux Winch: Payout of the main winch shall be through the use of the aux winch. For the main winch payin, the cables' dead weight shall be sufficient loading for the retrieval. Time of retrieval shall be recorded. For the aux winch, the shackles on the vehicle shall be used to load the cable during the retrieval. The cable dead weight pull shall be the no-load case. The full load main winch test shall be a 140,000 lb. static pull. The static pull shall be made against a static anchor until the winch stalls. At which point a load cell shall measure the pulling capacity. CSTA shall use three to four vehicles or their large Dynamometer as the anchor. The M88A1E1 has been designed for static, not dynamic testing. The static pull test shall be completed for 1, 2, and 3 layers of cable wrapped around the winch drum.

(d) Test Repetitions/Measurements: A minimum of three (3) repetitions of each test shall be made. Pressure, temperature, and flow recordings shall be made at five minute intervals. Should pressure readings exceed 5% of normal operating conditions, raw data recordings shall be made every minute.

(6) APG shall have the following test equipment available:

(a) Hydraulic test stand for the measuring and monitoring of temperatures and pressures.

(b) Five (5) each pressure transducers. They shall have digital readouts. The pressure transducers shall monitor pressures up to 5000 psi.

(c) Five (5) each fast response temperature sensors with digital read outs.

(d) A digital storage oscilloscope that is capable of monitoring the output of the pressure transducers.

(e) A strip chart readout for the temperature sensors. It shall have a minimum two (2) input channels, preferably five (5) if available.

(f) An x-y plotter to interface with the oscilloscope.

(g) 70,000 pounds of dead weight to be used during boom testing. A load cell that can read 70,000 lbs. shall be provided. Measurement instrumentation shall be included.

(h) CSTA's large dynamometer is required for the 140,000 lb. winch test. A load cell reading to 140,000 lb. is required. if the dynamometer is not available, alternate solutions would be two M60A3s and one M88A1 or two M88s with the plow blades down to simulate the same loading. The load cell shall include measurement instrumentation.

11. Test Scope:

a. Summary of Test Objectives:

(1) Determine to the best extent possible what hydraulic problems, if any, still exist in the five M88A1E1 prototype vehicles at APG. The testing will center on problems determined during PPT and documented in the cover letter or the TIRs.

(2) Become knowledgeable of the current systems installed in the M88A1E1. Accurate system installation drawings are not available as of 11 Jun 90.

(3) Determine any hydraulic problems that were not addressed in the TIRs. It is important to determine what these problems might be. Any additional problems found during the operational tests shall be documented for further evaluation and resolution.

(4) Determine solutions to hydraulic problems wherever possible.

b. Test Schedule: Operational testing of vehicle #4 at APG is scheduled to start the 9th of July and be completed by the 30 Aug 90.

12. Final Inspection: Upon completion of all testing, the vehicles tested shall be inspected to determine the condition of each vehicle and returned to Bldg. 420 for preparation for shipment.

13. Reporting:

a. Upon completion of the tests, TACOM, AMSTA-ZDS, shall prepare a detailed report describing the systems tested, test results, and any recommendations resulting from tests performed.

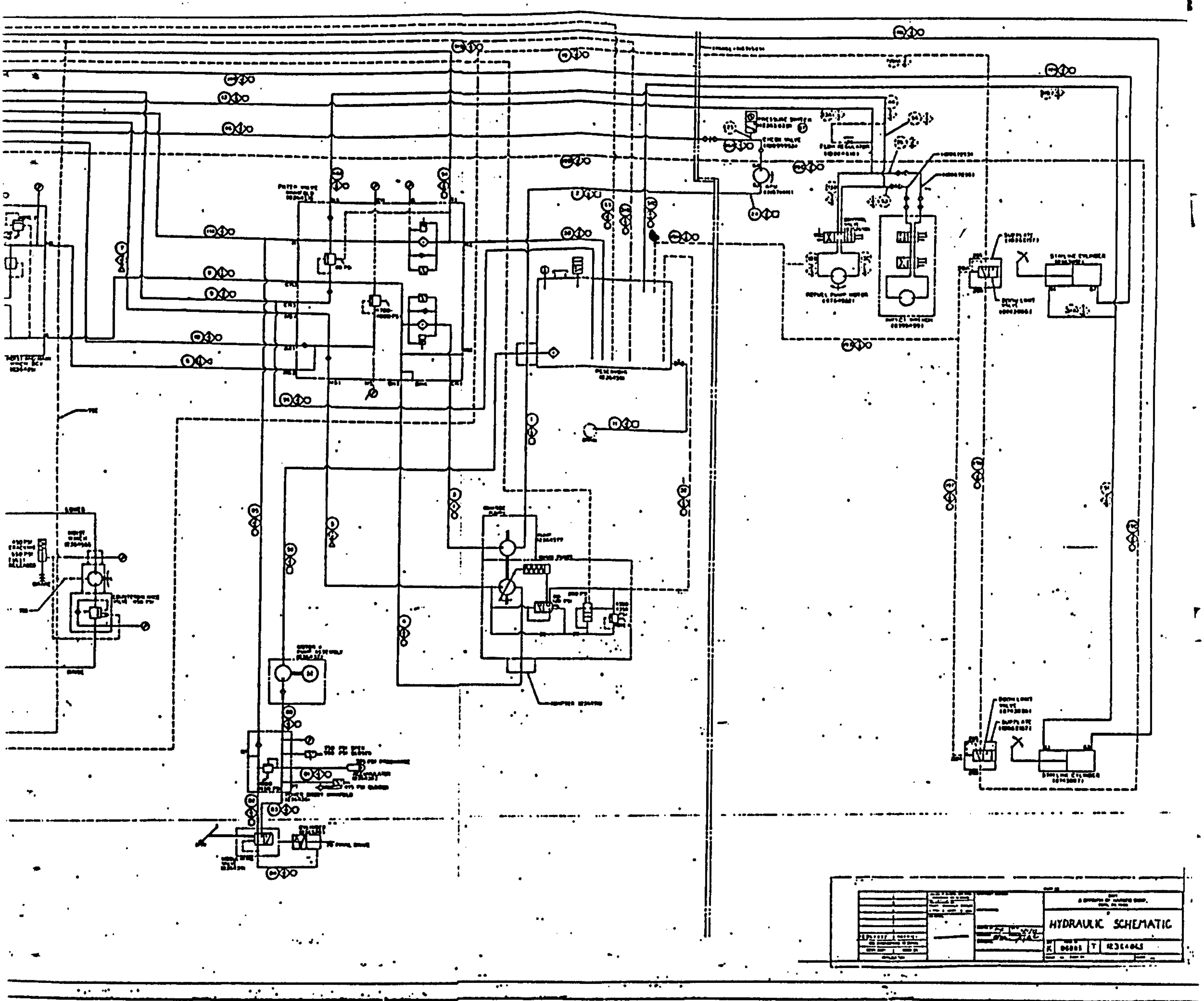
b. All pertinent information, major test incidents and/or failures, shall be immediately reported and confirmed by telephone to AMSTA-Z-IRV, phone # 1-313-574-8832.

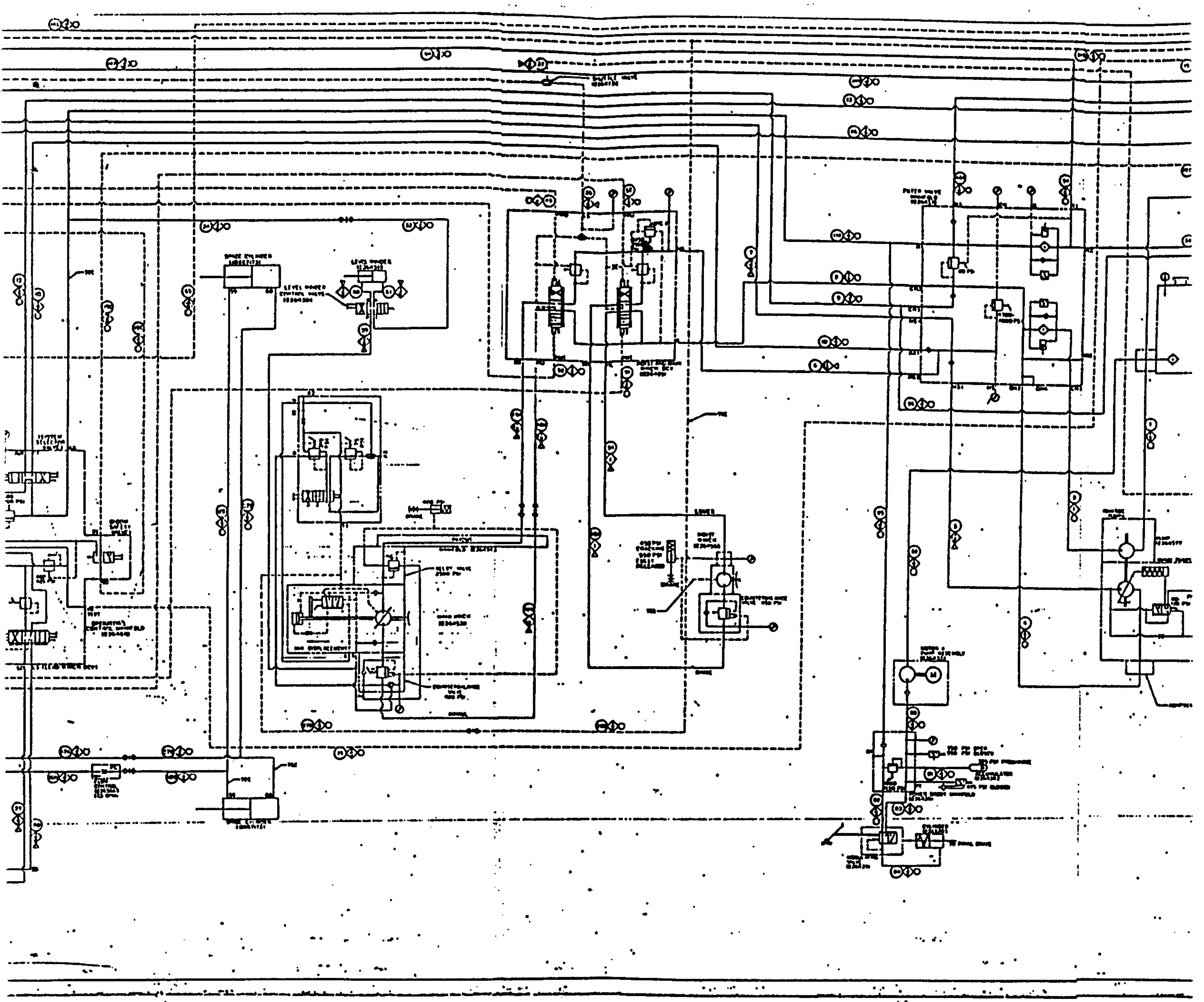
c. All testing results, TIRs and final test report shall be distributed only to the organization specifically listed below:

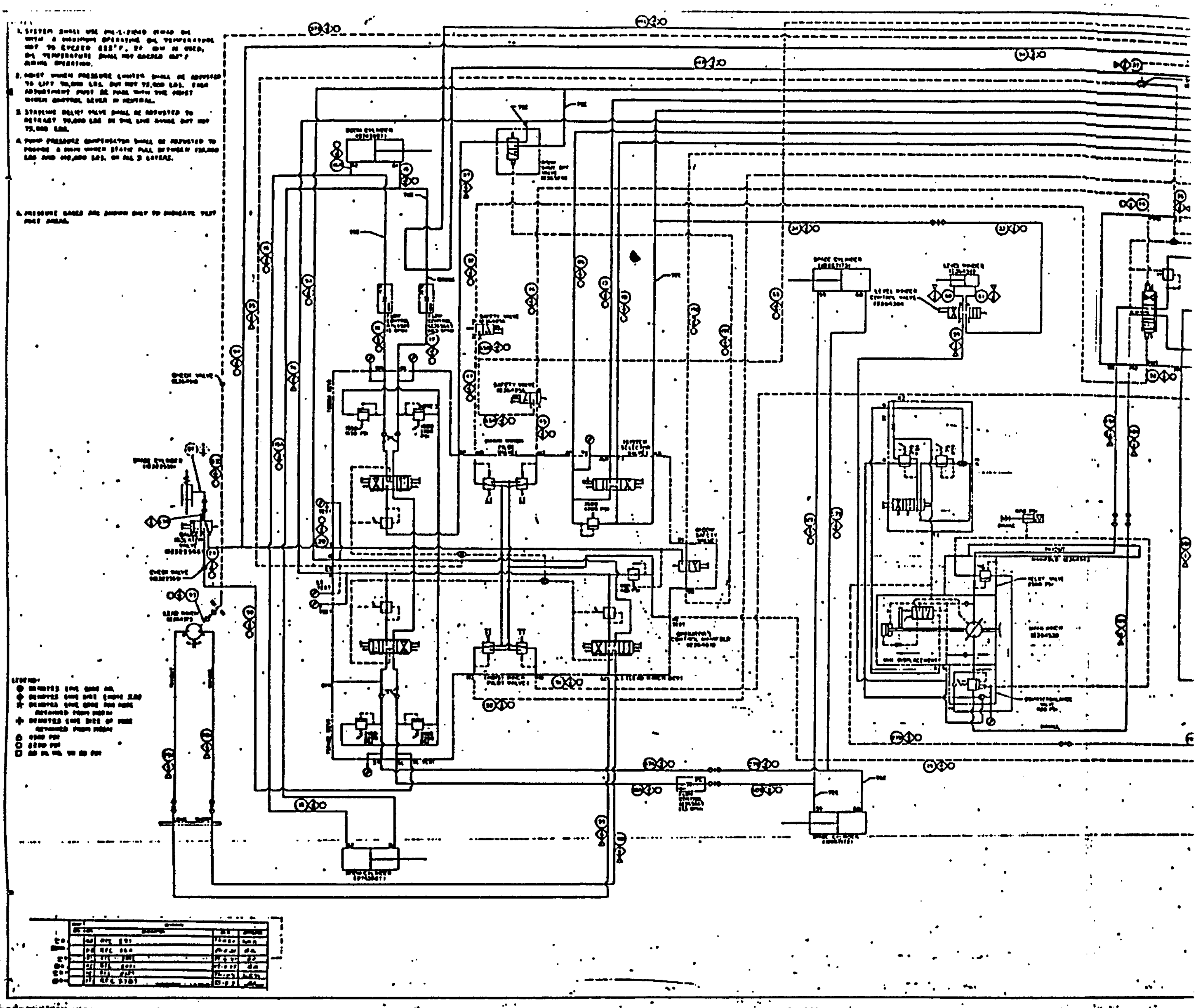
Commander
U.S. Army Tank-Automotive Command
AMSTA-Z-IRV
Warren, Mich 48397-5000
Attn: Mr. Ron Chapp

APPENDIX D

Hydraulic System Diagram


HYDRAULIC SCHEMATIC





APPENDIX E

Test Instrumentation

M88A1E1 HYDRAULIC SYSTEM TEST INSTRUMENTATION

PRESSURE TRANSDUCERS	RANGE	SERIAL NUMBER
Teledyne Taber model #2403		
Charge pump pressure CH-port on charge pump	200 PSI	895409
Main system pressure P-port on operator control manifold	5000 PSI	895135
Pump compensation pressure MS-port on side of CH-port	5000 PSI	895136
Winch motor comp pressure X-port on layer sense valve	500 PSI	854183
Main Directional Control Valve		
MO-port	5000 PSI	895138
MI-port	5000 PSI	895137

THERMOCOUPLES T type

Dipstick (main winch GC-port)

Hydraulic reservoir

Hoist winch return

Ambient

Inside

THERMOCOUPLES K type

Main directional control valve
MO-port
MI-port

Aux Winch in

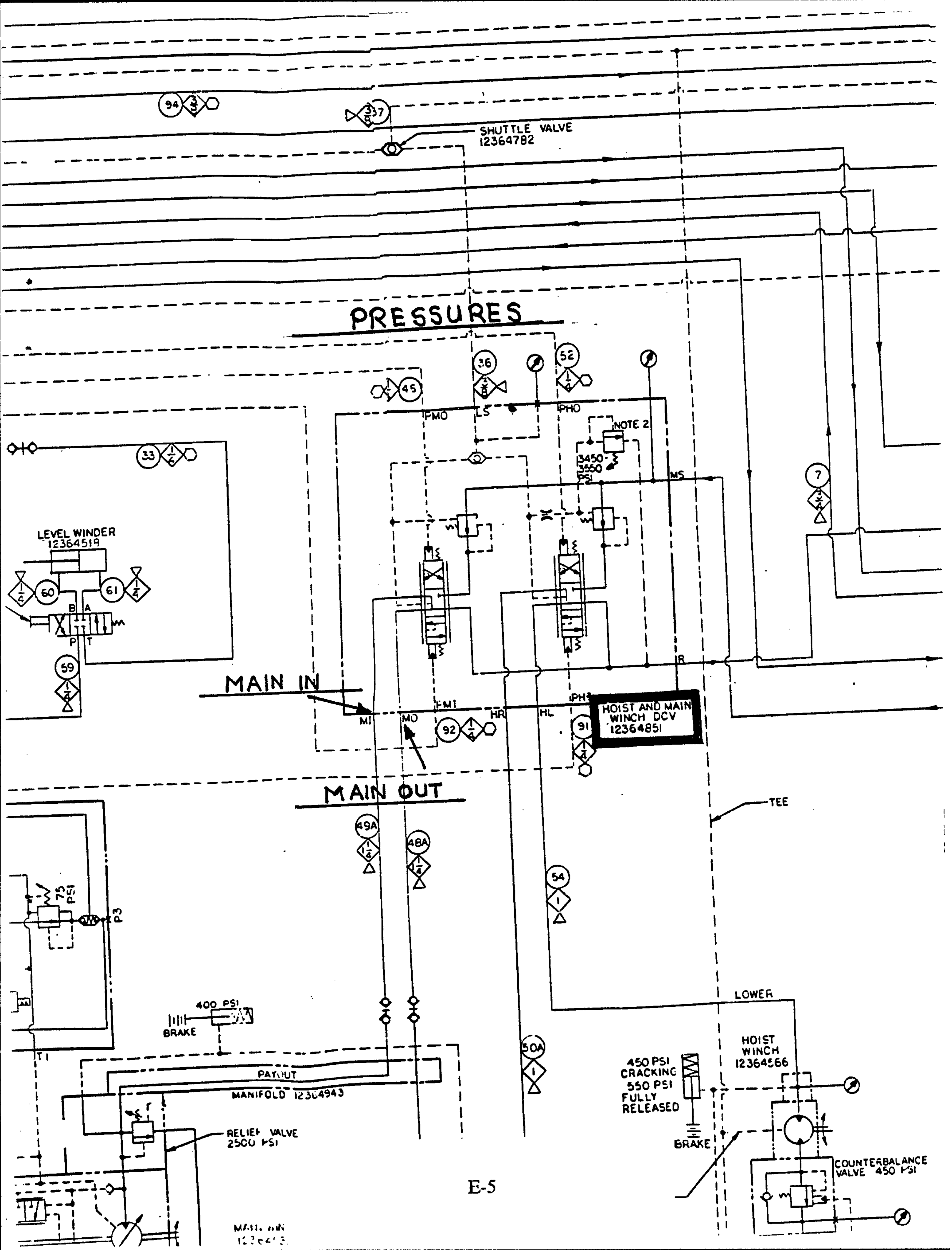
Aux winch out

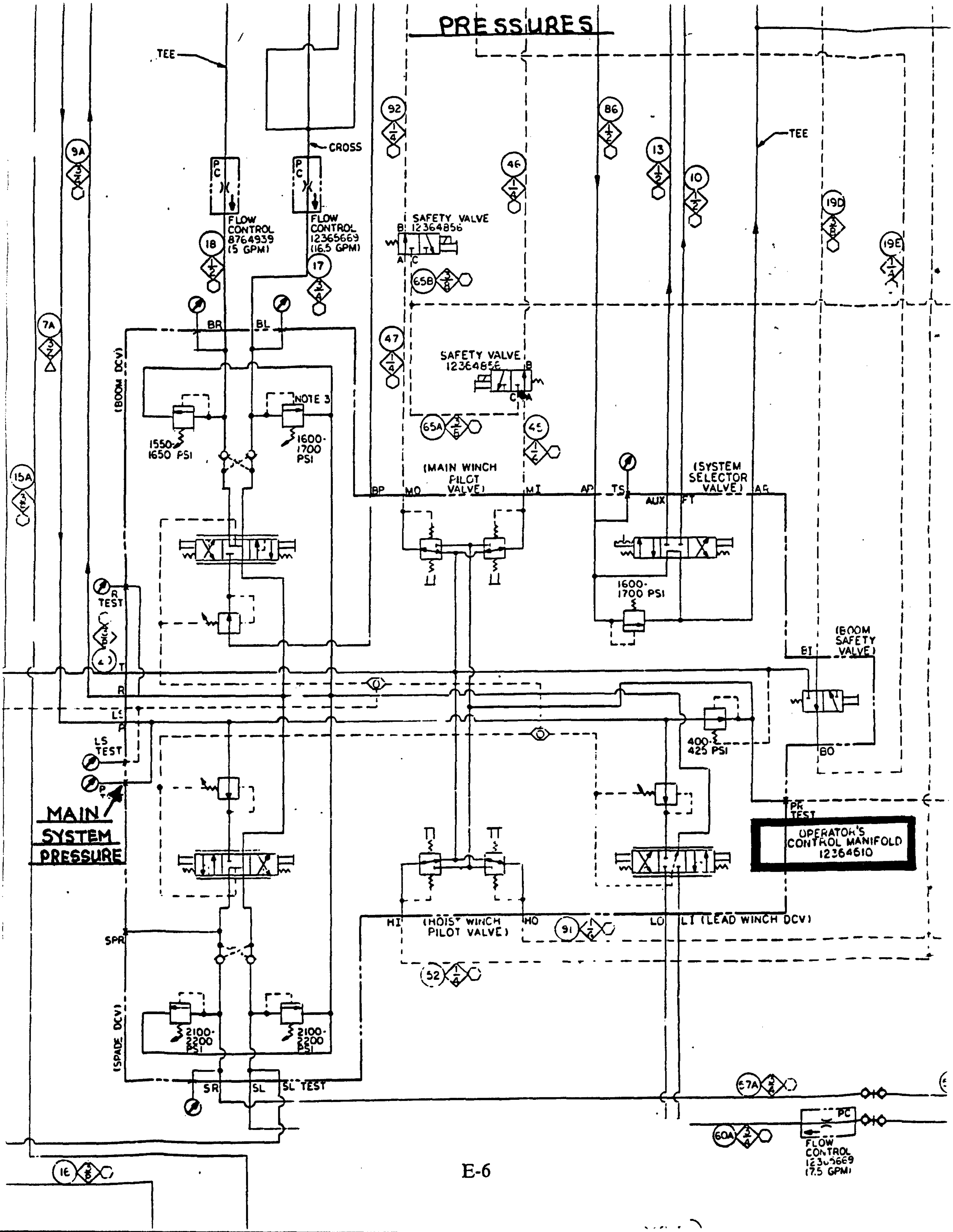
Hoist raise

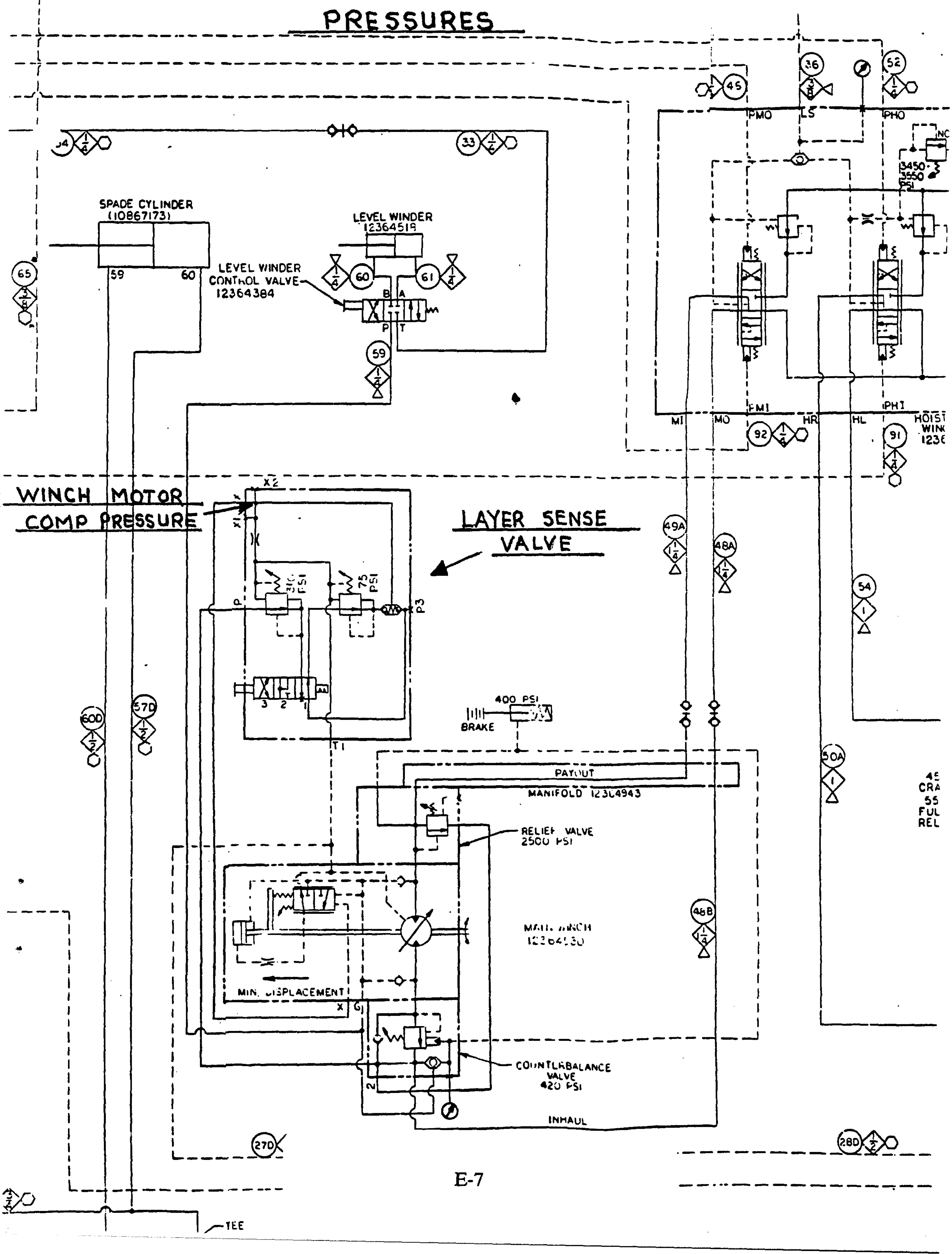
Hoist lower

Data recorder computer = Zenith laptop zwl-184-97

In house data acquisition unit used

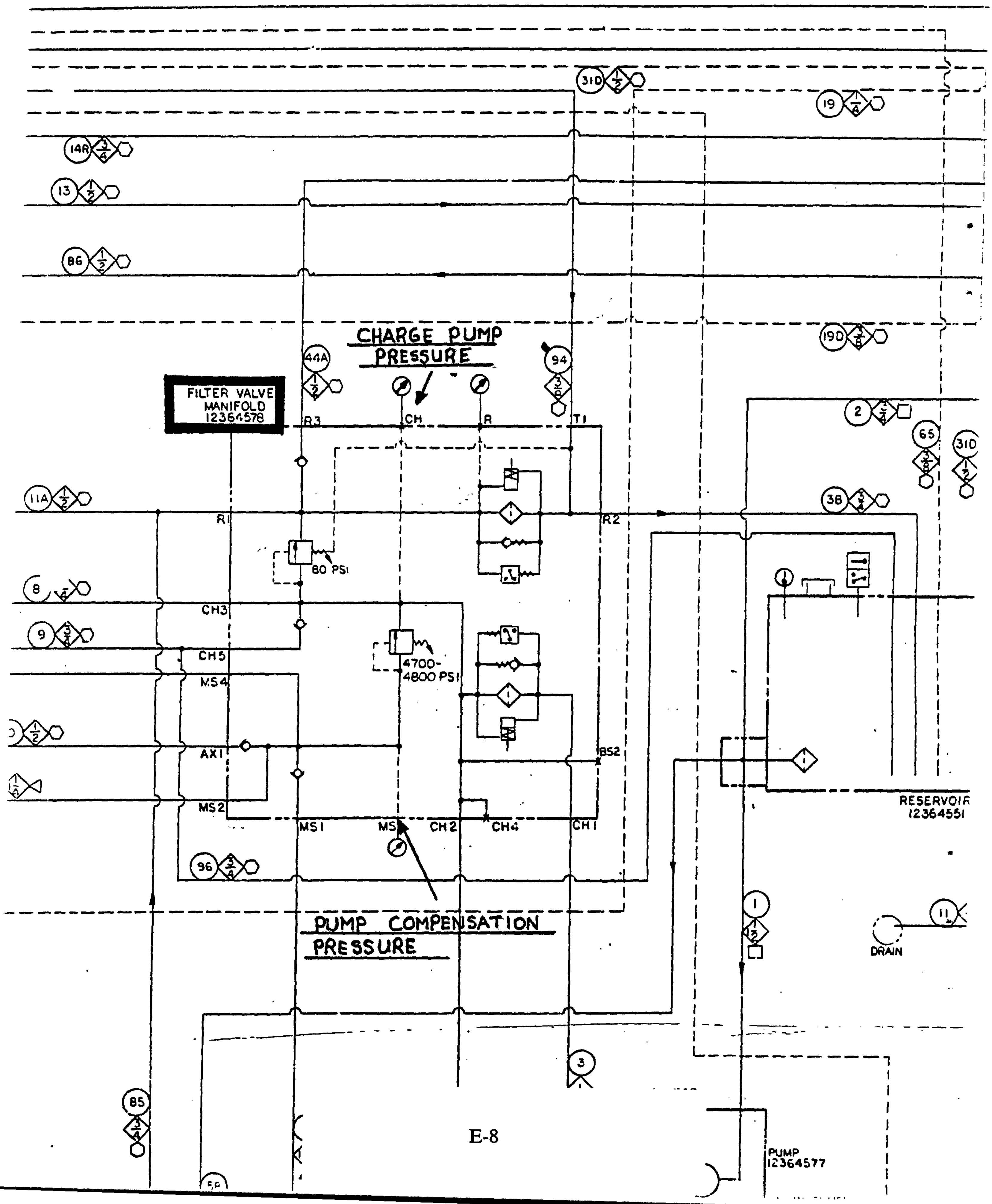

SHUTTLE VALVE
12364782
PRESSURES
PMO
LS
PHO
NOTE 2
3450-
3550
PSI
MS
LEVEL WINDER
12364519
B A
P T
MAIN IN
MI
MO
PMI
HR
HL
PH
R
HOIST AND MAIN
WINCH DCV
12364851
MAIN OUT
TEE
75
PSI
P3
400 PSI
BRAKE
T1
PAYOUT
MANIFOLD 12364943
RELIEF VALVE
2500 PSI
LOWER
HOIST
WINCH
12364566
450 PSI
CRACKING
550 PSI
FULLY
RELEASED
BRAKE
COUNTERBALANCE
VALVE 450 PSI


PRESSURES
TEE
CROSS
FLOW CONTROL 8764939 (5 GPM)
FLOW CONTROL 12365669 (16.5 GPM)
SAFETY VALVE 12364856
SAFETY VALVE 12364856
(BOOM DCV)
NOTE 3
1550-1650 PSI
1600-1700 PSI
(MAIN WINCH PILOT VALVE)
(SYSTEM SELECTOR VALVE)
1600-1700 PSI
(BOOM SAFETY VALVE)
R TEST
LS TEST
MAIN SYSTEM PRESSURE
400-425 PSI
PR TEST
OPERATOR'S CONTROL MANIFOLD 12364610
(HOIST WINCH PILOT VALVE)
(LEAD WINCH DCV)
(SPADE DCV)
2100-2200 PSI
2100-2200 PSI
SR
SL
SL TEST
SPR
FLOW CONTROL 12365669 (7.5 GPM)

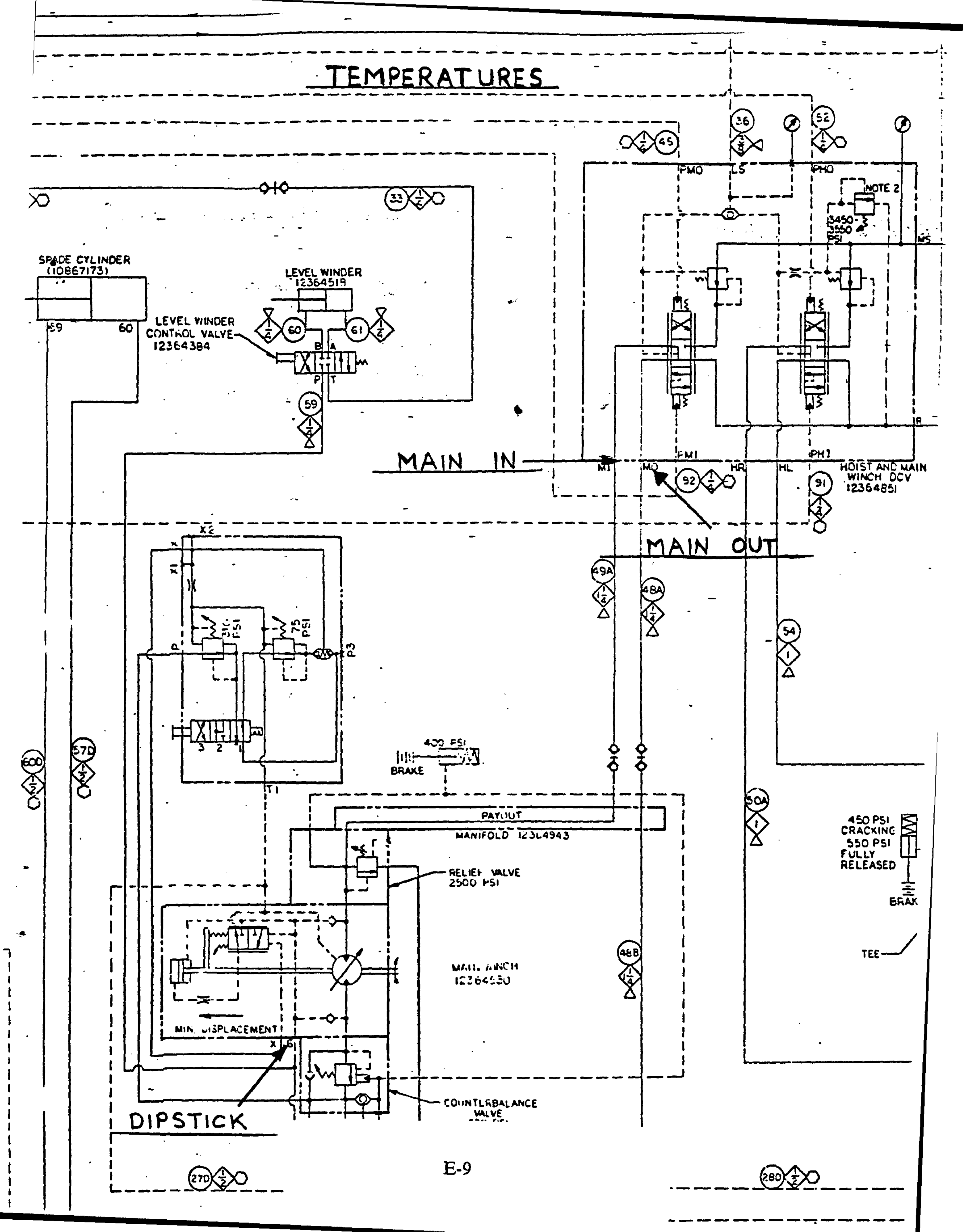


PRESSURES
SPADE CYLINDER
(10867173)
LEVEL WINDER
12364519
LEVEL WINDER
CONTROL VALVE
12364384
WINCH MOTOR
COMP PRESSURE
LAYER SENSE
VALVE
400 PSI
BRAKE
PAYOUT
MANIFOLD 12364943
RELIEF VALVE
2500 PSI
MIN. DISPLACEMENT
COUNTERBALANCE
VALVE
420 PSI
INHAUL
TEE

PRESSURES



CHARGE PUMP PRESSURE
FILTER VALVE MANIFOLD 12364578
R3
CH
R
T1
R1
R2
80 PSI
CH3
CH5
MS4
AX1
MS2
4700-4800 PSI
BS2
MS1
MS
CH2
CH4
CH1
PUMP COMPENSATION PRESSURE
RESERVOIR 12364551
DRAIN
PUMP 12364577



TEMPERATURES
SPADE CYLINDER
(10867173)
LEVEL WINDER
12364519
LEVEL WINDER
CONTROL VALVE
12364384
MAIN IN
MAIN OUT
HOIST AND MAIN
WINCH DCV
12364851
NOTE 2
3450-
3550
PSI
400 PSI
BRAKE
PAYOUT
MANIFOLD 12364943
RELIEF VALVE
2500 PSI
MIN. DISPLACEMENT
450 PSI
CRACKING
550 PSI
FULLY
RELEASED
TEE
COUNTERBALANCE
VALVE
DIPSTICK

APPENDIX F

Summer Test Chronology

SUMMER TEST CHRONOLOGY

Date	Test Event
8/16/90	Start of test, stall loads recorded
8/17/90	Main winch level wind breaks
8/22/90	Ground hop kit developed
8/22-9/12/90	Modifications to main winch level wind
9/13/90	Ground hop completed on P4 system leaks repaired P4 cable inspected prior to test one stall test successfully completed
9/16/90	Seven main winch pulls completed
9/17/90	Two main winch stall tests completed at 90% load
9/18/90	BMY reports P4 hydraulic system is plumbed differently than as shown in drawings submitted to TACOM
9/25/90	BMY/TACOM concur on the installation of the double pilot operated check valve for the level wind system
10/23/90	BMY travels to CSTA to make modifications to control valves for the main winch/hoist system
11/15/90	Modifications to main winch/hoist system complete
11/16/90	P4 ground hop completed
11/26/90	Hoist tested and found to be operational major winch failure
11/27/90	Main winch assembly of P2 installed into P4, rain cupola added to P4
12/4/90	Main winch test restarted
12/10/90	APG environmental shuts down test for hydraulic oil spill
12/11/90	First successful day of winch test completed

12/12/90	100% winch load completed
12/13/90	Hoist stall loads completed
12/20/90-3/1/91	Test halted due to Desert Storm priorities
3/6/91	Duty cycle testing started
3/7/91	PAL Filter Corp take oil samples during test
3/8/91	Three-pump design developed
3/26/91	Modifications made to P4: Three-pump system installed Rebuilt operators manifold installed
3/27/91	Three-pump modification complete, system ground hopped, vehicles P1, P2, P3, and P5 shipped to BMY
3/28/91	M88A1E1 test integration working group meeting (TIWG) held successful demonstration of P4 hydraulic capabilities demonstrated to TIWG members
4/1-3/91	Three-pump duty cycle tests completed
4/4/91	Test halted test instrumentation removed from P4
4/11/91	P4 shipped to BMY

APPENDIX G

Summer Test Incidents

SUMMER TEST INCIDENTS

DATE	INCIDENT(I)/RESOLUTION(R)
8/17/90	**I:** Level wind cam shaft angle bracket sheared at flanges
	R: Gusset added to bracket for rigidity; installed new cam shaft case hardened to a Rockwell C hardness of 45
	I: Cam follower shoulder bolt failed
	R: New shoulder bolt installed
9/13/90	**I:** Many system hydraulic oil leaks observed
	R: Improperly installed connectors reinstalled
9/17/90	**I:** Trumpet movement noted at system shutdown, movement causes shoulder bolt fracture and fleet angle bracket to shear
	R: Installation of double-pilot-operated check valve (DPOCV) eliminates cylinder drift, new bolt and bracket installed
9/24/90	**I:** TACOM/BMY hydraulic system review indicates hoist system to be disconnected due to winch modifications during original FY89 prototype testing
	R: Hoist system reconnected by: a. Replumbing of main winch/hoist directional control valve b. Load sense lines rerouted
11/16/90	**I:** Air in level wind cylinder; level wind cylinder lagged behind position of diamond screw; fleet angle switch bracket sheared off
	R: System bled of air and bracket replaced
11/17/90	**I:** Cam follower riding in extreme positions, fleet angle switches tripping
	R: The cam was moved closer to the cylinder control valve thus giving a full open position to the valve and eliminating the extreme positions

11/26/90	I: Main winch cable birds-nested
	R: Operators error caused failure, cable replaced
12/4/90	I: Minor hosing oil leaks
	R: Main winch/hoist DCV shuttle valve not installed properly, valve was correctly installed and leaks were eliminated
	I: DPOCV connector hose broke
	R: New hose installed
12/5/90	I: Data recording error
	R: Velocity meter replaced
12/5/90	I: Test was restarted; at third layer inhaul, system pressure dropped rapidly
	R: Second DPOCV hose broke; new hose installed
12/13/90	I: Erratic behavior of hoist
	R: Cold temperatures affect the operation of the hoist, warming the fluid above 80°F eliminated the erratic behavior
3/6/91	I: BMY hook block failure
	R: Bushing design of BMY hook block was ineffective, Johnson roller bearing block installed
3/27/91	I: System oil leaks
	R: All leaks successfully resolved

APPENDIX H

Test Data

Test Data

Data for the M88A1E1 hydraulic summer test was gathered over the period 16 Aug 90 to 2 Apr 91 by the CSTA test personnel. Eighty-one (81) total test runs were made. The data was collected in the following sequence:

a. Stall data

b. Baseline winching (Runs 1-53):

1.) 50% (70,000lb. load) winch inhaul

2.) 75% (105,000lb. load) winch inhaul

3.) 90% (126,000lb. load) winch inhaul

4.) 100% (140,000lb. load) winch inhaul

5.) 110% (154,000lb. load) winch inhaul

c. Baseline hoisting (Runs 54-63)

d. Duty Cycle test, two-pump system (Runs 64-71)

e. Duty Cycle test, three-pump system (Runs 72-81)

All test data gathered is given in Tables 1 - 3.

Test runs 1-63 covered baseline test data. The operation of the vehicle was being tested during these runs. Test runs 64-81 represent duty cycle testing. In Section 5 of this report modifications were made to the hydraulic system. These modifications are included in Runs 64-81. Also in Section 5, reference is made to the installation of a three-pump system. This three-pump system was installed for test runs 72-81.

For each test run made, the raw test data was taken by CSTA and plotted in graph form by computer. Temperatures and pressures were plotted versus time for each test run. A composite graph for temperatures and pressures was also given for each run. The following graphs were plotted:

a. Temperatures vs. Time for:

1.) Main winch

2.) Hydraulic reservoir

3.) Hoist winch

4.) Ambient

5.) Inside (crew compartment)

6.) Main out (winch)

7.) Main in (winch)

8.) Aux in (at aux winch)

9.) Aux out (at aux winch)

10.) Hoist raise

11.) Hoist lower

b. Pressure vs. Time for:

1.) Main system

2.) Main in

3.) Pump compensation

4.) Main out

5.) Charge pump

6.) Winch motor comp

Volumes of graph data were plotted for the 81 runs. Graph data for runs 1-63 represent data for the nonmodified system and are not pertinent to this report. For this reason, the graph data is not included in this report. Runs 64-71 represent data for the two pump modified system. The graph data for one run was typical of the data for the other seven runs and, therefore, the data for only one run, number 71, is presented. In a like manner, run 81 was included to represent the three-pump modified system. Runs 71 and 81 have been chosen as a comparison to exhibit the differences in pressures and temperatures between the two-pump and three-pump systems. The benefit gained by installing the three pump-system was determined by this comparison to be a 30% reduction in hydraulic oil heat gain.

Figure 1 is presented as reference in describing the pressure fluctuations for a typical duty cycle. The pressure spikes are labeled to describe the cycle operation. The duty cycle followed the following sequence of events:

a. spade down

b. boom up

c. hook block lowered

d. first hoist

e. hoist lower

f. second hoist

g. hoist lower

h. third hoist

i. hoist lower

j. boom up

k. spade up

l. aux out

m. aux pull

n. aux in

o. first layer 140,000 lb. main winch pull

p. second layer main winch pull

r. third layer main winch pull

s. spade up

t. charge pump (idle pressure)

Figure 2 depicts the break down of the duty cycle into the lift, aux winch, and main winch cycles.

TABLE 1. M88A1E1 SUMMER TEST DATA

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
#1 8/16/90										
Full Payout							1800	4300		
#2 8/16/90										
Stall Bare Drum							1820	4250	173,990	
Stall Bare Drum							1826	4250	178,490	
2nd Layer							1822	4225	165,300	
2nd Layer							1821	4225	166,490	
3rd Layer							1818	4200	146,890	
3rd Layer							1816	4200	154,550	
#3 8/17/90										
Payout	0956	1006		86	121	35	1785	1800	13,570	
#4 8/17/90										
Bare Drum	1105			121			1758	3600	57,010	16.05
2nd Layer										16.73
3rd Layer		1115	10		153	28			55,350	17.75
#5 8/17/90										
Payout	1120	1134	14	155	168	13	1758	1600	16,250	
#6 8/17/90										
Main Inhaul	1503	1514	11	115	152	37	1800	3800	72,000	
#7 9/17/90										
Main Outhaul	0855	0904	9	72	105	33	1804	556		

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
#8 9/17/90										
Main Inhaul	1001	1008	7	133	155	22	1790	3550	43,890	
Bare Drum										44.50
2nd Layer										42.50
3rd Layer										38.20
#9 9/17/90										
Main Outhaul	1051	1057	6	142	155	22	1815	2000		
#10 9/17/90										
Main Inhaul	1105	1114	9	165	195	30	1785	3400	75,250	
Bare Drum										31.00
2nd Layer										31.20
3rd Layer										27.30
#11 9/17/90										
Main Outhaul	1321	1327		154	176	22	1795	1650	21,820	
#12 9/17/90										
Main Inhaul	1330	1349		175	215	40	1785	4250	110,570	
Bare Drum										17.50
2nd Layer										18.60
3rd Layer										15.20
#13 9/17/90										
Main Outhaul	1406	1413		211	226	15	1785	1450	31,420	
Bare Drum										90.77
2nd Layer										87.34
3rd Layer										73.62

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
#14 9/17/90										
Cooling	1424	1515		223	196	-27	810	600	0	
#15 9/17/90										
Main Inhaul	1517	1522		196	203			2700	17,750	
Bare Drum										62.90
2nd Layer										71.00
3rd Layer										64.20
#16 9/18/90										
Main Outhaul	0854	0905	11	87	113	26		1700	19,550	
Bare Drum										84.27
2nd Layer										76.92
3rd Layer										72.82
#17 9/18/90										
Main Inhaul	0910	0925	15	113	159	46	1760	4200	105,270	
Bare Drum										23.80
2nd Layer										24.10
3rd Layer										21.70
#18 9/18/90										
Main Inhaul	0930	0935	5	155	179	24		1700	14,620	
Bare Drum										79.58
2nd Layer										79.47
3rd Layer										71.01
#19 9/18/90										
Main Inhaul	0946	1004	18	175	214	39	1760	4200	126,480	
Bare Drum										21.90

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
2nd Layer										20.70
3rd Layer										Stall
#20 9/18/90										
Main Outhaul	1008	1016	8	210	216	6	1780	1700	20,000	
Level Wind Failed										
#21 12/5/90										
Stall Bare Drum	1352	1407		117	122		1800	4328	155,550	
2nd Layer Stall								4358	154,560	
#22 12/5/90										
2nd Layer Stall	1408	1417		121	130		1800	4390	159,000	
#23 12/5/90										
3rd Layer Stall	1546	1553		89	110		1800	4390	149,995	
#24 12/10/90										
Payout	1033	1041					1800			
3rd Layer								1900	42,850	
2nd Layer								3500	43,340	
Bare Drum									72,850	
#25 12/10/90										
Payin	1050	1057		79	131		1800			
Bare Drum								3000	20,000	
2nd Layer								3000	23,000	
3rd Layer								2500	25,000	
#26 12/11/90										

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Payout	1022	1032		59	82					
#27 12/11/90										
Payin	1037	1059		84	126		1800			
Bare Drum Stall				88				4300	155,220	31.66
2nd Layer Stall								4200	151,680	
3rd Layer Stall								4275	142,690	
#28 12/11/90										
Payout	1103	1109		126	144		1800	1900	12,380	
#29 12/11/90										
Payin	1110	1120		141	167		1800	3400		
Bare Drum										32.57
2nd Layer										32.73
3rd Layer										29.76
#30 12/11/90										
Payout	1122	1128		166	180			1900		
Bare Drum									8,690	91.46
2nd Layer									12,900	32.73
3rd Layer									16,000	81.52
#31 12/11/90										
Payin	1132	1146		179	214			4200		
Bare Drum									116,890	22.09
2nd Layer									116,080	21.42
3rd Layer									114,490	20.13
#32 12/11/90										

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Payout	1430	1436		177	161		1800	2000	8,040	
#33 12/11/90										
Payin	1449	1512		165	187		1800			
Bare Stall				159				4300	159,730	
2nd Stall				171				4250	157,540	24.86
3rd Layer				181				4200	148,500	22.67
#34 12/11/90										
Payout	1520	1525		185	198			2000	14,780	
#35 12/11/90										
Stall	1528	1538		196	204					
Bare Drum				197				4225	156,100	
2nd Layer				198				4200	149,340	
3rd Layer				202				4238	143,840	
#36 12/12/90										
Payout	1004	1012		71	82		1800	1900	8,230	
#37 12/12/90										
Payin	1020	1033		86	118		1800			
Bare Drum								4000	100,230	24.60
2nd Layer								3950	99,850	24.67
3rd Layer								3500	10,125	22.95
#38 12/12/90										
Payout	1038	1042		120	137		1800	1900	7,800	
#39 12/12/90										

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Payin	1046	1056		137	163					
Bare Drum								4200	126,740	21.77
2nd Layer								3600	69,680	30.91
3rd Layer								3100	68,130	28.57
#40 12/12/90										
Payout	1059	1103		162	177		1800	1800	14,025	
#41 12/12/90										
Payin	1107	1118		174	196		1800			
Bare Drum								4200	126,320	21.42
2nd Layer								3100	68,910	29.25
3rd Layer								3100	68,520	27.84
#42 12/12/90										
Payout	1120	1125		195	208		1800	1700	16,400	
#43 12/12/90										
Payin	1422	1438		151	162		1800			
Bare Drum								4250	140,390	19.90
2nd Layer								3550	68,890	31.10
3rd Layer								3200	68,830	28.41
#44 12/12/90										
Payout	1440	1445		162	174		1800	1800	13,490	
#45 12/12/90										
Payin	1447	1458		174	198					
Bare Drum								4200	140,680	19.50
2nd Layer								3200	68,610	30.90

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
3rd Layer								3000	65,450	27.03
#46 12/13/90										
Payout	0913	0921		74	86		1800	1800	16,830	
#47 12/13/90										
Payin	0923	0935		88	114					
Bare Drum								4300	154,580	12.00
2nd Layer								3500	68,840	31.32
3rd Layer								3250	68,200	27.52
#48 12/13/90										
Payout	0937	0943		116	133		1800	1800	9,680	
#49 12/13/90										
Payin	0945	0956		132	160					
Bare Drum								4300	153,900	14.06
2nd Layer								3500	66,740	30.86
3rd Layer								3200	63,800	28.35
#50 12/13/90										
Payout	0958	1003		160	176			2000	4,500	
#51 12/13/90										
Payin	1007	1017		174	195					
Bare Drum								4000	102,800	24.97
2nd Layer								3200	66,720	139.20
3rd Layer								3300	63,800	27.33
#52 12/17/90										

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Aux Payout	1356	1407		47	58					
#53 12/17/90										
Aux Payin	1436	1500		69	88		800			
Bare Drum									6,420	
2nd Layer									6,230	
3rd Layer									5,930	
4th Layer									5,730	
5th Layer									5,530	
6th Layer									5,020	
#54 3/3/91										
Hoist	1102	1104		163	164		1800	4250	77,500	
Hoist Stalled on 3rd Layer										
#55 3/3/91										
Hoist	1107	1108		163	164		850	4250	77,000	
Hoist Stalled on 3rd Layer										
#56 3/3/91										
Hoist (3)	1116	1130		163	160		850	4250	80,000	
									88,600	
									88,500	
#57 3/3/91										
Hoist (2)	1133	1136		160	160		1800	4250	88,600	
									89,500	
#58 3/3/91										
Hoist	1334	1340		142	140		850	3500	70,950	

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Test Load ---> 9.31 seconds for 10 feet (lowering)										
#59 3/3/91										
Hoist	1348	1354		139	143		1800	3950	70,950	
Test Load ---> 8.76 seconds for 10 feet										
#60 3/3/91										
Checkout	1356	1407		143	144		850		70,950	
#61 3/5/91										
Spade Lower	1012	1022		57	101		1800	3950	Stall	
#62 3/5/91										
Hoist	1045	1100	15	102	124	22	1800	3950		
Duty Cycle										
3 Hoists	1335	1422	45	111	169	58				
Aux Payout	1337	1342		111	107		1800	900		
3rd Layer	1344	1348		107	113			2500		
2nd Layer	1351	1354		118	130			2500		
Bare Drum	1359	1400		130	135			4200		
Aux Payin	1400	1406		135	140					
Bare Drum	1411	1414		140	149		1800	4200	143,000	20.69
2nd Layer	1414	1417		149	156			3750	70,000	31.31
3rd Layer	1417	1422		156	169		1800	3250	70,000	28.22

TABLE 2. M88A1E1 SUMMER TEST DATA (Duty Cycle Sequence, 2 Pumps)

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
#64 3/6/91										
Hoist Cycle	0922	1015	53	75	167	92				
Lift 1	0931			82	91	9		3785	70,950	66.4[a]
Lift 2				91	98	7		3865	70,950	68.2
Lift 3		0941	10	98	106	8		3865	70,950	68.8
Aux Winch	0946	0958	12	110	138	28		985/4245		
Main Winch										
Bare Drum	0958			138	144	6		4245	140,000	18.9
2nd Layer				144	156	12		3635	68,000	32.5
3rd Layer		1011	13	156	167	11		3260	68,000	28.9
#65 3/6/91										
Hoist Cycle	1022	1115	53	166	202	37				
Lift 1	1028			158	161	3		3750	70,950	69.2[b]
Lift 2				161	168	7		3940	70,950	69.2
Lift 3		1038	10	168	171	3		4015	70,950	68.2[c]
Aux Winch	1042	1059	17	170	177	7		760/4245		
Main Winch										
Bare Drum	1101			178	183	5		4280	140,000	17.2
2nd Layer				183	192	9		3525	68,000	30.3
3rd Layer		1112	11	192	203	11		3260	68,000	28.2
#66 3/7/91										
Hoist Cycle	1309	1357	48	80	164	84				
Lift 1	1319			90	99	9		3635	70,950	67.0[d]
Lift 2				99	105	6		3710	70,950	67.6
Lift 3		1325	6	105	112	7		3600	70,950	67.0
Aux Winch	1332	1345	13	117	141	24		680/4280		

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Main Winch										
2nd Layer	1346			141	145	4		4320	140,000	15.1
3rd Layer		1355	9	154	164	10		3260	70,000	29.0
#67 3/7/91										
Hoist Cycle	1416	1456	40	158	210	52				
Lift 1	1422			153	158	5		3600	70,950	67.6[e]
Lift 2				158	166	8		3710	70,950	68.6
Lift 3		1428	6	166	172	6		3710	70,950	68.0
Aux Winch	1434	1443	9	171	186	15		680/4240		
Main Winch										
2nd Layer	1445			187	200	13		4240	140,000	17.8
3rd Layer		1454	9	200	210	10		3180	68,000	29.5
#68 3/7/91										
Hoist Cycle	1503	1551	48	204	242	38				
Lift 1	1510			194	198	4		3790	70,950	69.3[f]
Lift 2				198	205	7		3865	70,950	68.6
Lift 3		1517	7	205	212	7		4165	70,950	61.0[g]
Aux Winch	1524	1533	9	208	222	14		680/4280		
Main Winch										
2nd Layer	1537			222	233	10		4240	140,000	14.9[h]
3rd Layer		1547	10	233	242	9		3180	65,000	27.4
#69 3/8/91										
Hoist Cycle	0857	0945	48	85	173	88				
Lift 1	0905			84	94	10		3750	70,950	66.7[i]
Lift 2				94	106	6		3750	70,950	66.7
Lift 3		0912	7	100	143	6		3750	70,950	67.5
Aux Winch	0917	0931	14	106	143	37		680/4280		

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Main Winch										
Bare Drum	0931			143	153	10		4250	141,000	18.8
2nd Layer				153	161	8		3600	68,700	31.4
3rd Layer		0944	13	161	173	12		3400	68,700	28.2
#70 3/8/91										
Hoist Cycle	0959	1040	41	166	221	55				
Lift 1	1004			161	162	1		3750	70,950	67.6[j]
Lift 2				162	169	7		3790	70,950	66.6
Lift 3		1010	6	169	175	6		3790	70,950	67.5
Aux Winch	1015	1027	12	175	197	22		610/4240		
Main Winch										
Bare Drum	1028			196	205	9		4250	140,000	19.5
2nd Layer				205	212	7		3640	69,700	29.6
3rd Layer		1039	11	212	222	10		3180	69,700	27.4
#71 3/8/91										
Hoist Cycle	1114	1156	42	202	246	44				
Lift 1	1121			193	193	0		3780	70,950	67.3[k]
Lift 2				193	198	5		3860	70,950	67.8
Lift 3		1127	6	198	203	5		3940	70,950	66.7
Aux Winch	1132	1142	10	202	222	20		640/4240		
Main Winch										
Bare Drum	1143			222	230	8		4240	140,000	18.9
2nd Layer				230	237	7		3410	68,000	29.0[l]
3rd Layer		1154	11	237	246	9		3110	68,000	27.1

[a] Average line speed as the test weight was lowered after three lifts (65.9 ft/min) utilizing an average main hydraulic system pressure of 1365 psi.

[b] Average line speed as the test weight was lowered after three lifts (69.3 ft/min) utilizing an average main hydraulic system pressure of 1365 psi.

[c] During this lift the hoist winch stalled as the test weight was within five feet of the top boom sheaves; this was caused by the hook block and boom sheaves being out of alignment and/or overheating.

[d] Average line speed as the test weight was lowered after three lifts (64.8 ft/min) utilizing an average main hydraulic system pressure of 1365 psi.

[e] Average line speed as the test weight was lowered after three lifts (69.2 ft/min) utilizing an average main hydraulic system pressure of 1365 psi.

[f] Average line speed as the test weight was lowered after three lifts (69.2 ft/min) utilizing an average main hydraulic system pressure of 1365 psi.

[g] During this lift the hoist winch stalled as the test weight was within five feet of the top boom sheaves; this may have been caused by high hydraulic oil temperatures.

[h] Winch stalling at approximately 142,000 lbs.

[i] Average line speed as the test weight was lowered after three lifts (65.4 ft/min) utilizing an average main hydraulic system pressure of 1360 psi.

[j] Average line speed as the test weight was lowered after three lifts (67.9 ft/min) utilizing an average main hydraulic system pressure of 1360 psi.

[k] Average line speed as the test weight was lowered after three lifts (67.3 ft/min) utilizing an average main hydraulic system pressure of 1360 psi.

[l] The hydraulic system high temperature warning light came on when the oil in the reservoir reached 235°F.

TABLE 3. M88A1E1 SUMMER TEST DATA (Duty Cycle Sequence, 3 Pumps)

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
#73 3/28/91										
Hoist Cycle	1100	1157	57	85	158	73				
Lift 1	1108			86	89	3		3360	70,950	64.0[a]
Lift 2				91	95	4		3390	70,950	64.2
Lift 3		1120	12	99	103	4		3690	70,950	66.4
Aux Winch	1127	1145	18	110	133	23		900/3485		
Main Winch										
Bare Drum	1146			125	149	24		4400	140,000	19.1
2nd Layer				149	153	4		3660	48,000	40.2
3rd Layer		1157	11	153	158	5		3210	48,000	35.6
#74 3/28/91										
Hoist Cycle	1205	1251	46	155	211	56				
Lift 1	1210			159	168	9		3490	70,950	66.4[b]
Lift 2				168	173	5		3180	70,950	65.8
Lift 3		1218	8	175	184	9		3260	70,950	65.9
Aux Winch	1227	1238	11	160	176	16		980/3490		
Main Winch										
Bare Drum	1239			170	188	18		4390	140,000	19.3
2nd Layer				188	201	13		3480	46,000	40.7
3rd Layer		1215	12	201	211	10		3110	46,000	36.5
#75 3/28/91										
Hoist Cycle	1504	1555	51	176	199	37				
Lift 1	1509			163	160	-3		2880	70,950	65.7[c]
Lift 2				161	165	4		4620	70,950	Stall
Lift 3		1515	6	165	174	9		4390	70,950	Stall
Aux Winch	1520	1541	21	156	178	22		910/3480		

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Main Winch										
Bare Drum	1543			172	192	20		4620	140,000	17.8
2nd Layer				192	201	9		3480	68,000	37.0
3rd Layer		1552	9	201	213	12		3030	68,000	31.5
#77 3/29/91										
Total Duty Cycle	1010	1039	29	63	123	60				
Aux Winch Cycle	1011	1022	11	63	87	24		1140/3480		
Out	1011	1015	4	63	66	3		1520		
Pull	1015	1020	5	66	76	10		3480		
In	1020	1022	2	76	87	11		1140		
Main Winch										
Bare Drum	1024			85	99	14		4240	140,000	22.1
2nd Layer				99	107	8		3410	56,000	36.6
3rd Layer		1037	13	107	123	16		3030	47,000	33.8
#78 3/29/91										
Total Duty Cycle	1041	1106	25	120	163	43				
Aux Winch Cycle	1043	1054	11	120	115	-5		950/3480		
Out	1043	1046	3	120	119	-1		1210		
Pull	1047	1052	5	119	126	7		3480		
In	1052	1054	2	126	115	-11		950		
Main Winch										
Bare Drum	1054			115	142	27		4390	141,000	20.2
2nd Layer				142	146	4		3480	48,000	40.1
3rd Layer		1106	12	146	163	19		3030	47,000	35.6
#79 3/29/91										
Total Duty Cycle	1112	1134	22	149	192	43				

Run#/Date	Time Start	Time Stop	Total Min.	Reservoir Temp, °F Start	Reservoir Temp, °F Stop	Reservoir Temp, °F Rise	Engine RPM	System Pressure PSI	Load LBS	Line Speed ft/min
Aux Winch Cycle	1112	1123	11	149	162	13		910/3480		
Out	1112	1116	4	149	151	2		1290		
Pull	1116	1120	4	151	158	7		3480		
In	1120	1123	3	158	162	4		910		
Main Winch										
Bare Drum	1124			162	172	10		4280	141,000	19.7
2nd Layer				172	179	7		3330	47,000	42.1
3rd Layer		1134	10	179	192	11		3030	43,000	35.4
#80 3/29/91										
Total Duty Cycle	1137	1159	22	178	216	38				
Aux Winch Cycle	1139	1148	9	178	187	9		1140/3450		
Out	1139	1142	3	178	178	0		1140	1,140	
Pull	1142	1146	4	178	184	6		3450		
In	1146	1148	2	184	187	3		1290		
Main Winch										
Bare Drum	1148			187	203	16		4470	140,000	19.3
2nd Layer				203	209	6		3330	46,000	41.6
3rd Layer		1159	11	209	216	7		3000	45,000	34.7
#81 3/29/91										
Total Duty Cycle	1203	1223	20	191	236	45				
Aux Winch Cycle	1203	1211	8	191	193	2		980/3370		
Out	1203	1206	3	191	178	-13		980		
Pull	1206	1210	4	178	196	18		3370		
In	1210	1211	1	196	193	-3		1210		
Main Winch										
Bare Drum	1212			192	220	28		4550	140,000	18.2
2nd Layer				220	229	9		3330	46,000	40.3
3rd Layer		1223	11	229	236	7		3000	45,000	34.1

[a] Average line speed as the test weight was lowered after three lifts (65.4 ft/min) utilizing an average main hydraulic system pressure of 1150 psi.

[b] Average line speed as the test weight was lowered after three lifts (67.9 ft/min) utilizing an average main hydraulic system pressure of 910 psi.

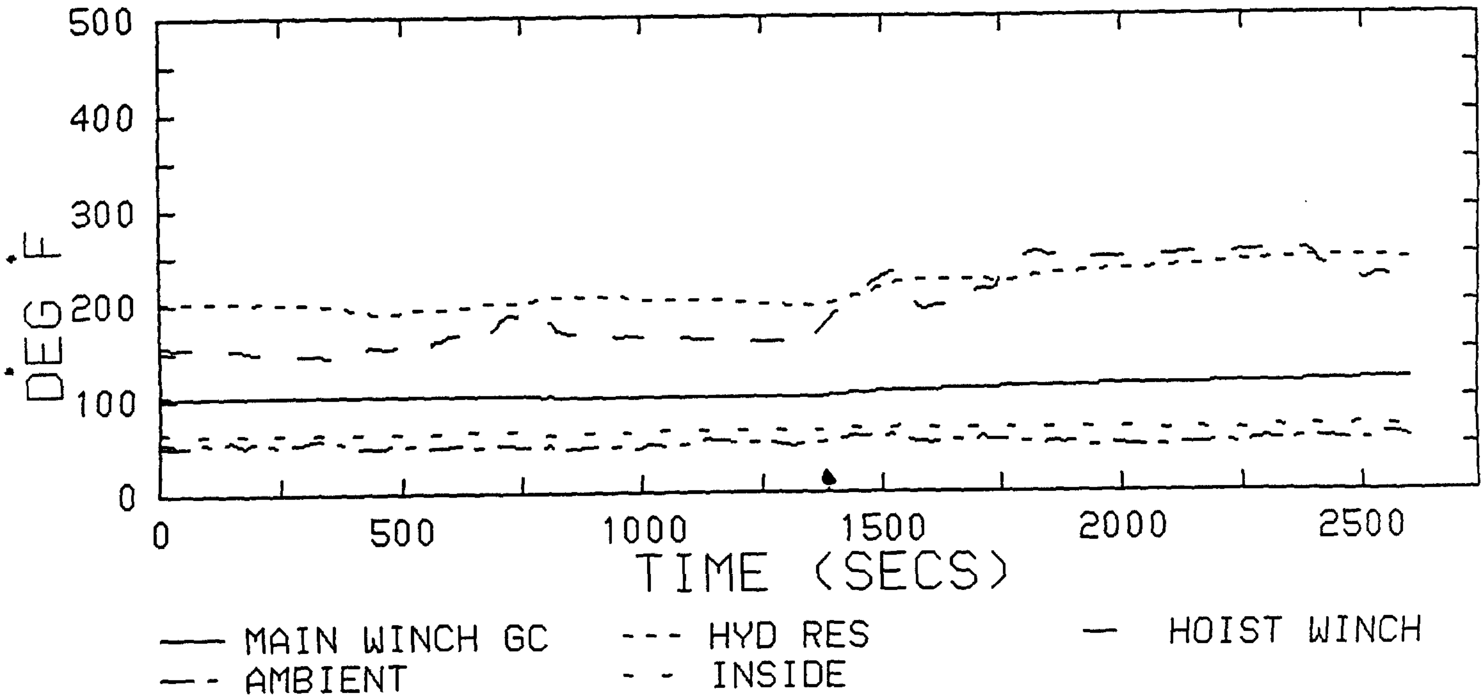
[c] Average line speed as the test weight was lowered after three lifts (67.3 ft/min) utilizing an average main hydraulic system pressure of 900 psi.

TRIAL #81

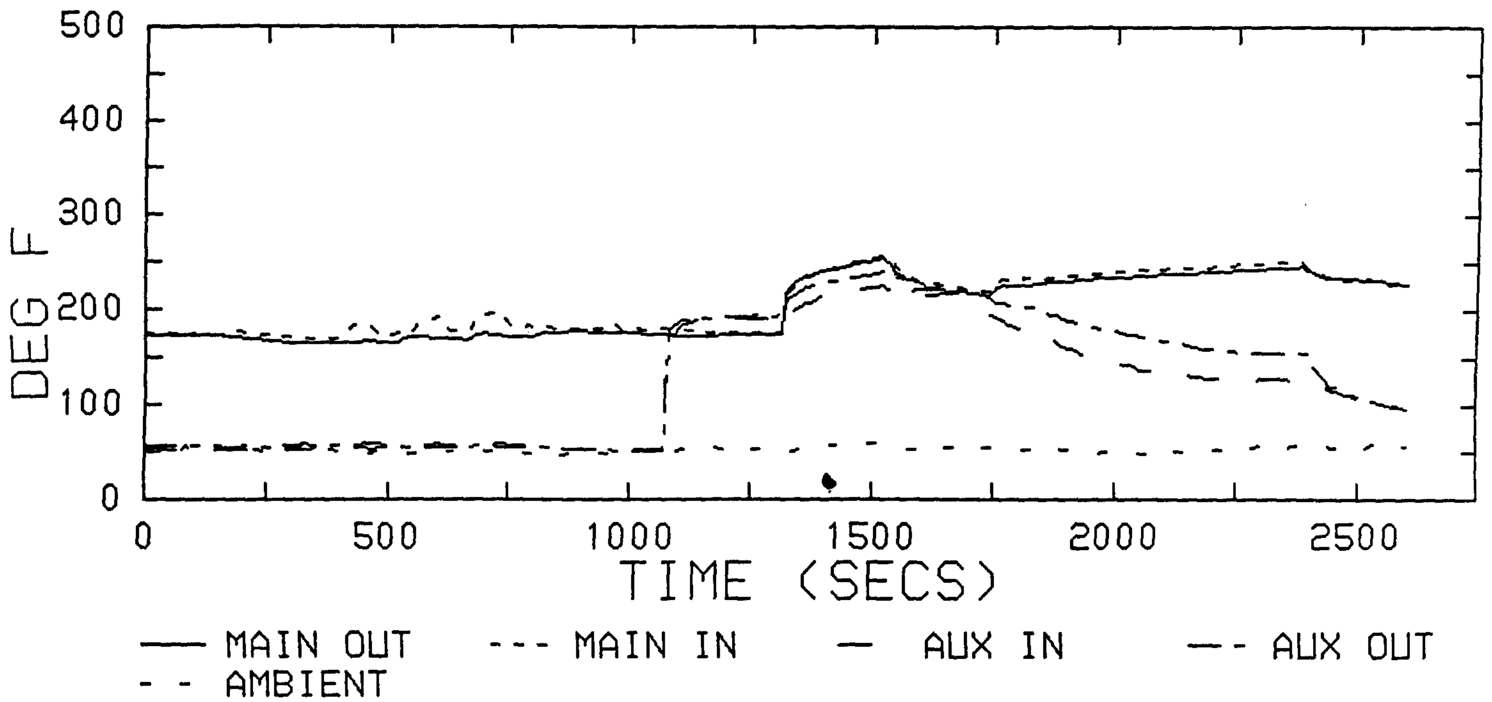
M88A1E1 (vehicle P4)

29 March 91

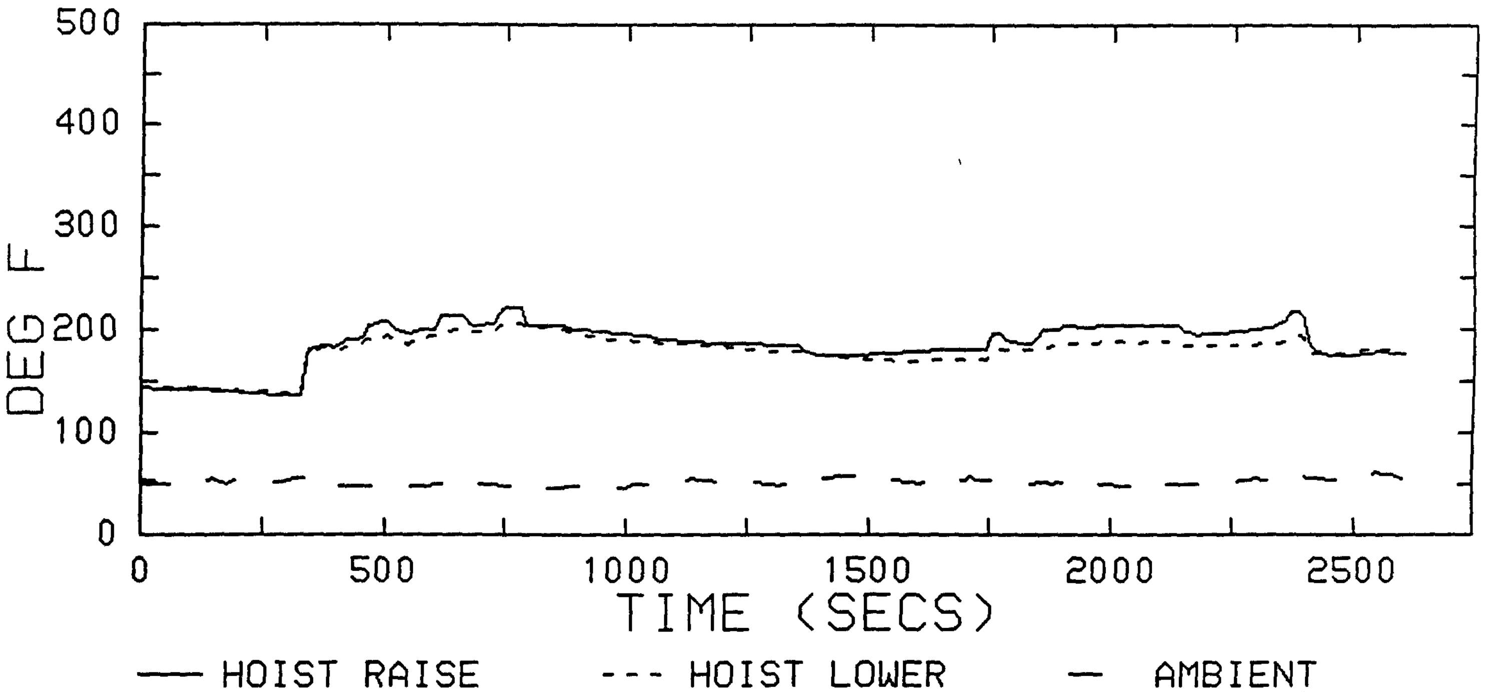
M88A1E1 P4 TEMPERATURES

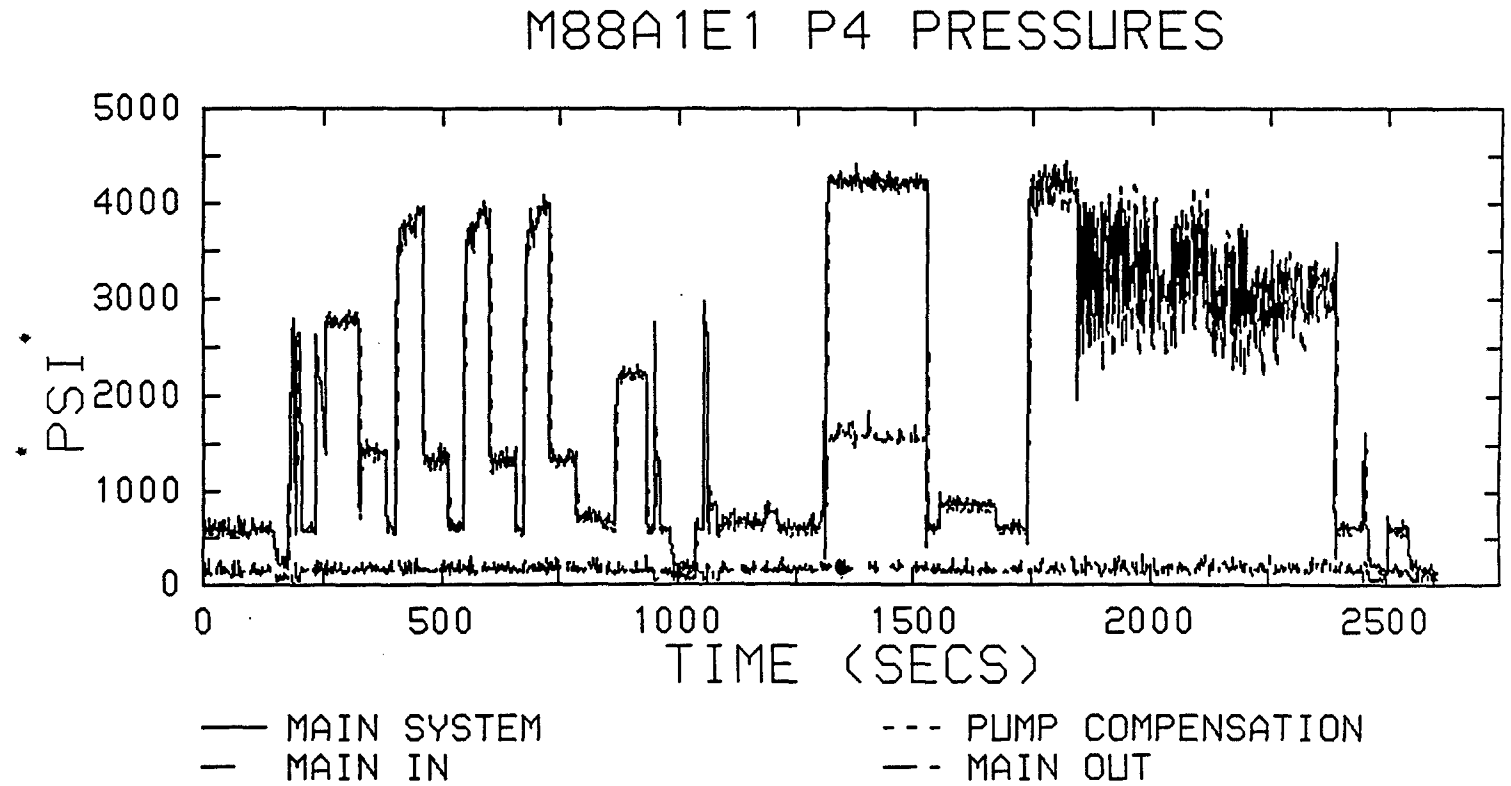


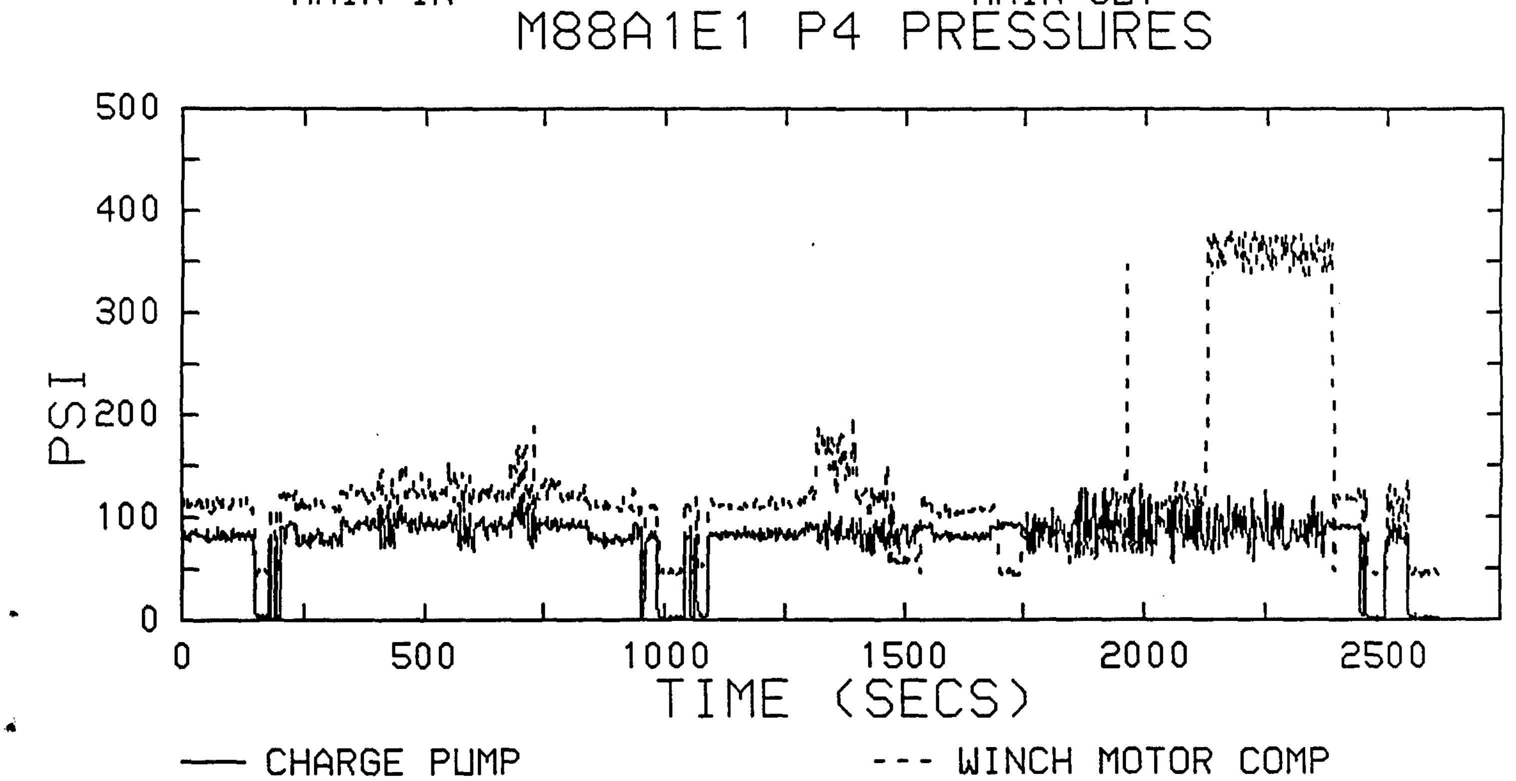
M88A1E1 P4 TEMPERATURES

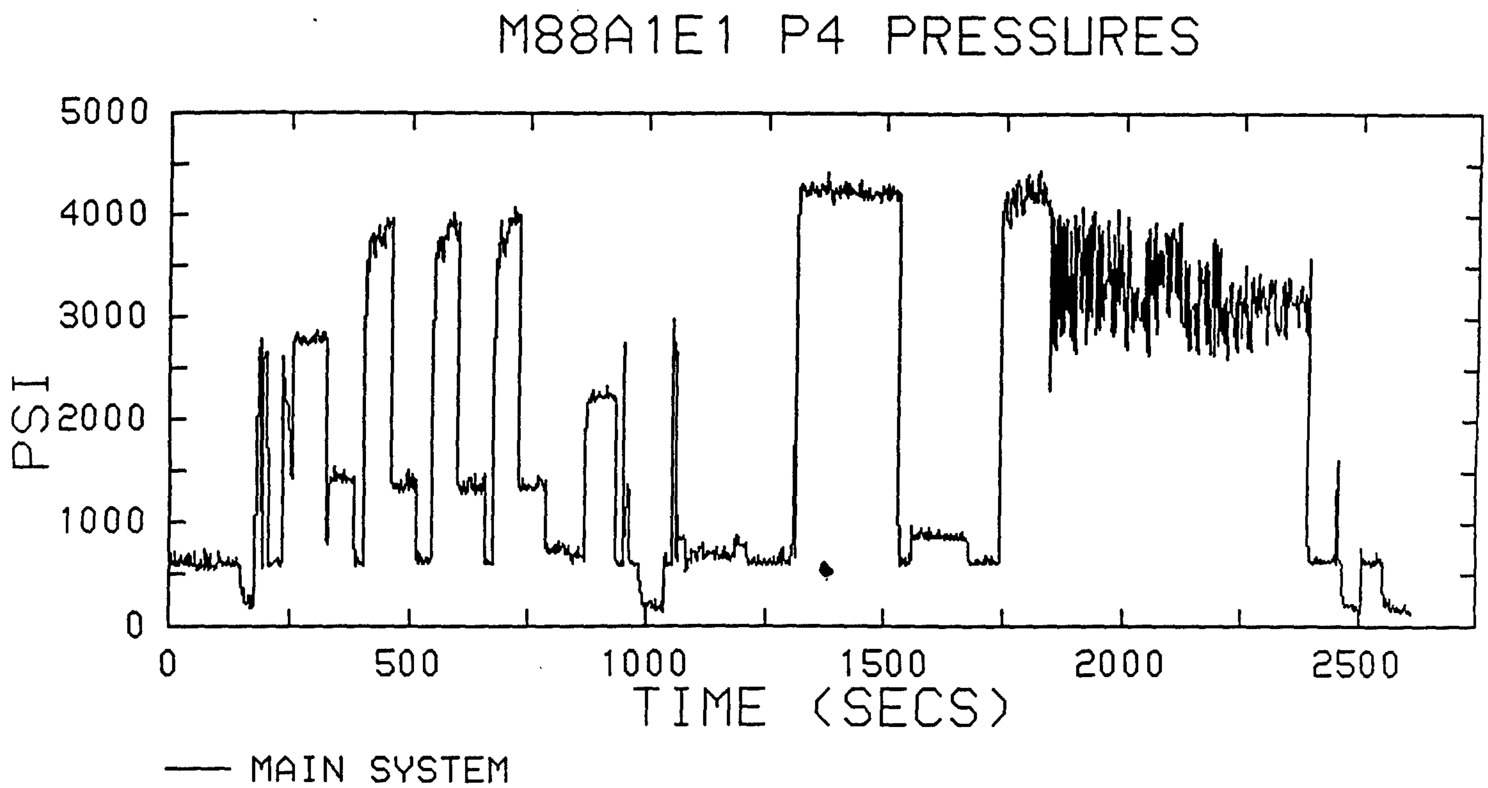


M88A1E1 P4 TEMPERATURES

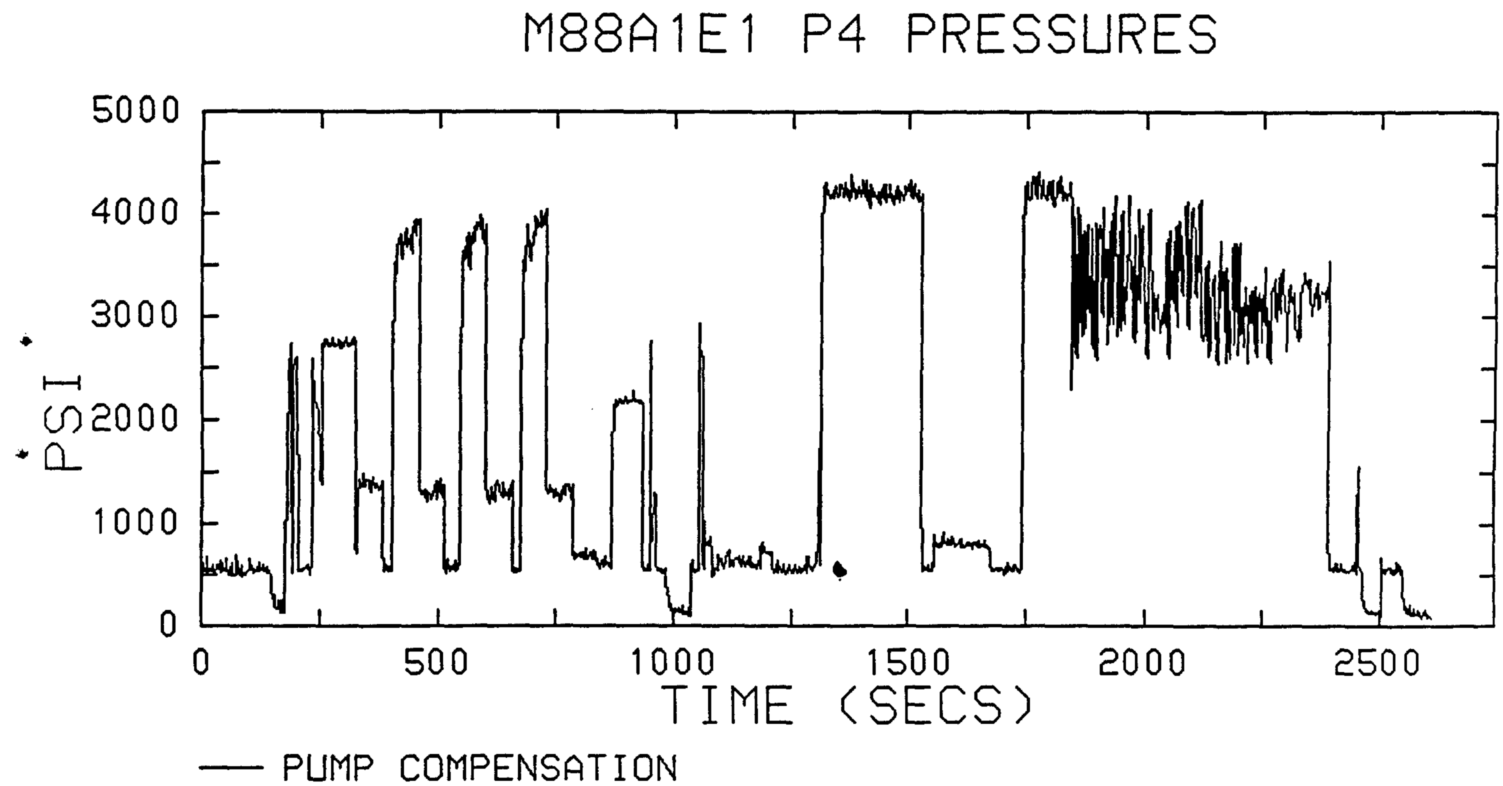


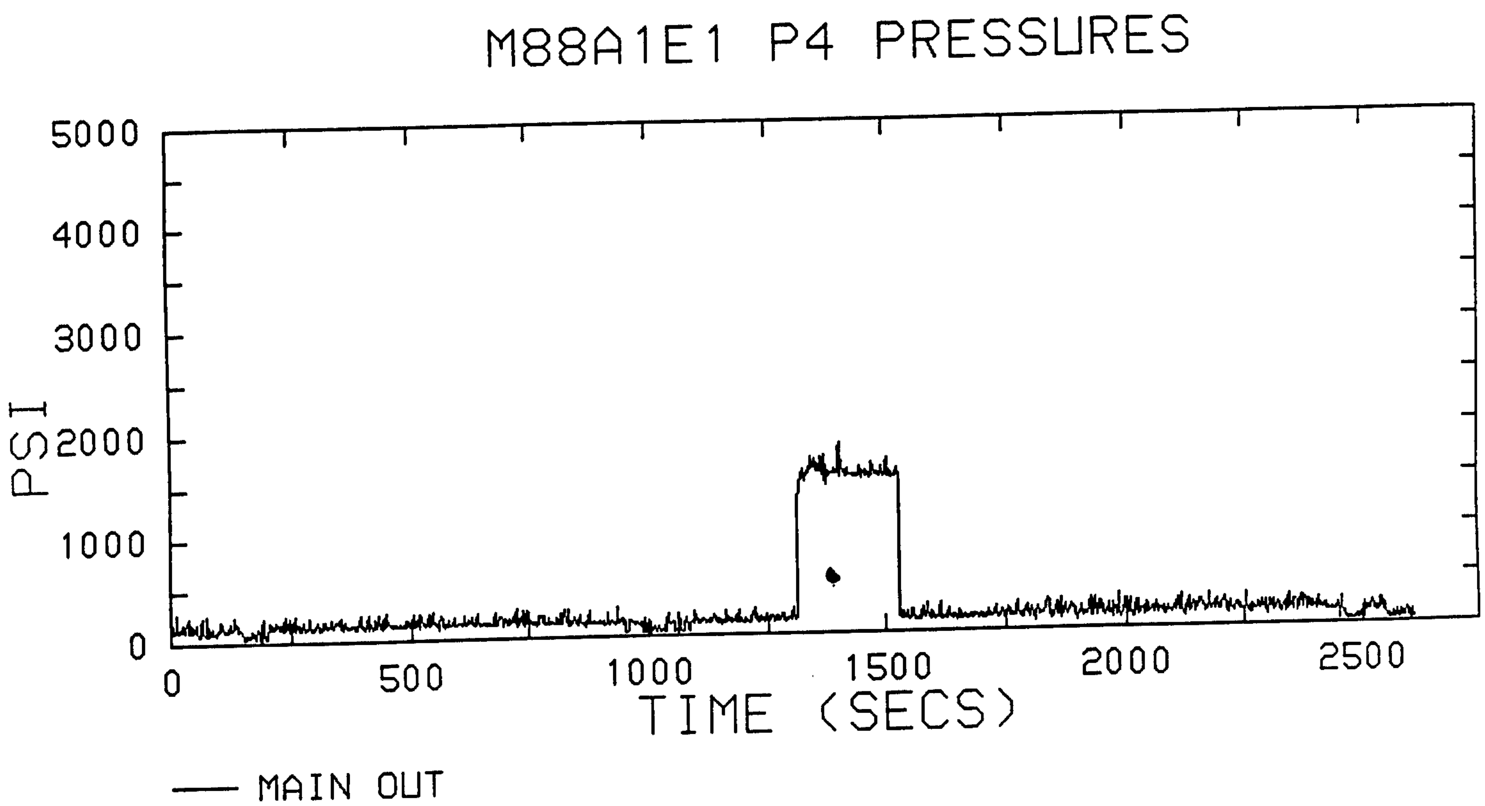

M88A1E1 P4 PRESSURES
5000
4000
3000
2000
1000
0
PSI
0
500
1000
1500
2000
2500
TIME (SECS)
MAIN SYSTEM
MAIN IN
PUMP COMPENSATION
MAIN OUT


M88A1E1 P4 PRESSURES
500
400
300
200
100
0
PSI
0
500
1000
1500
2000
2500
TIME (SECS)
CHARGE PUMP
WINCH MOTOR COMP

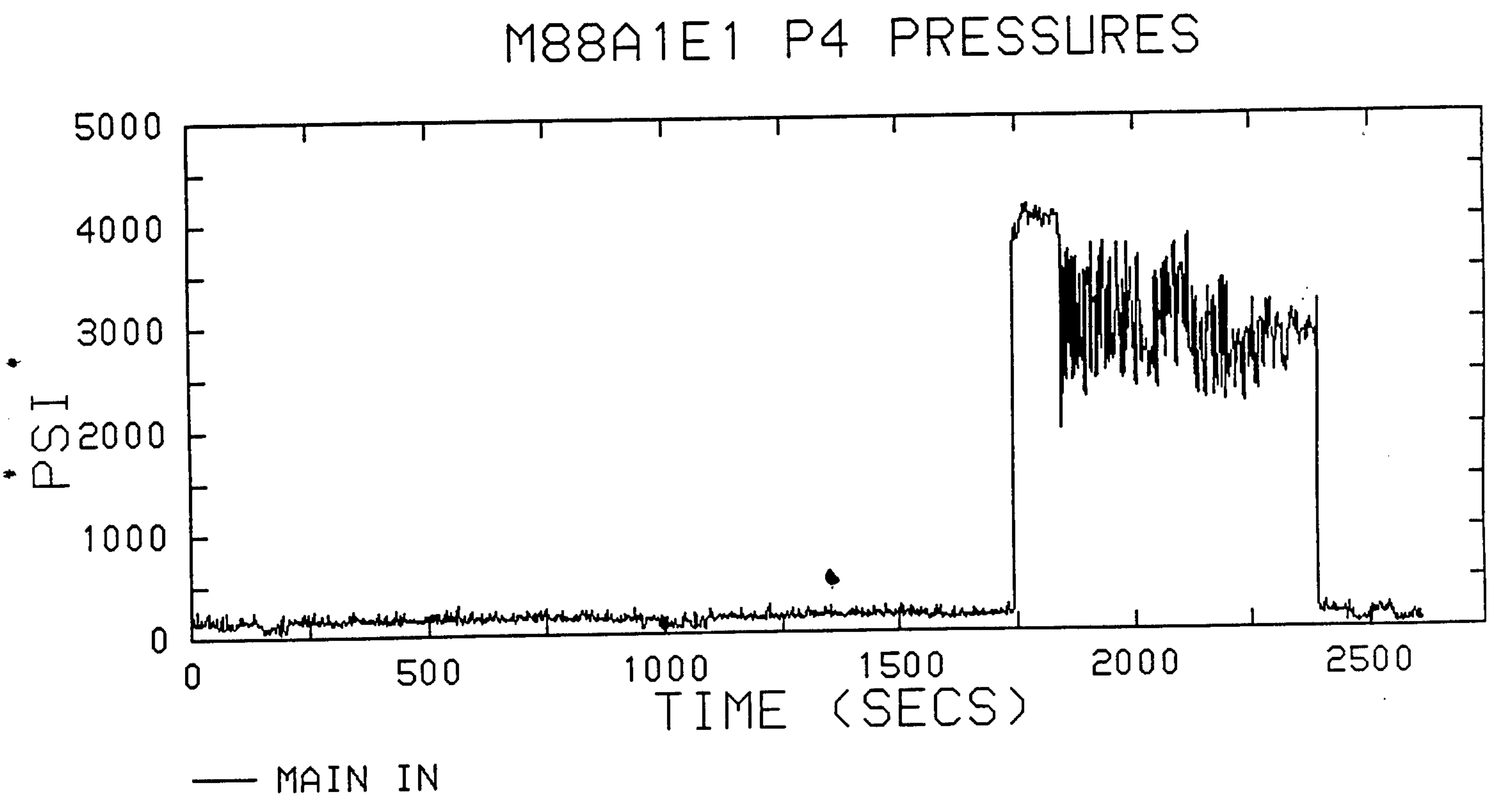


M88A1E1 P4 PRESSURES
PSI
5000
4000
3000
2000
1000
0
0
500
1000
1500
2000
2500
TIME (SECS)
MAIN SYSTEM

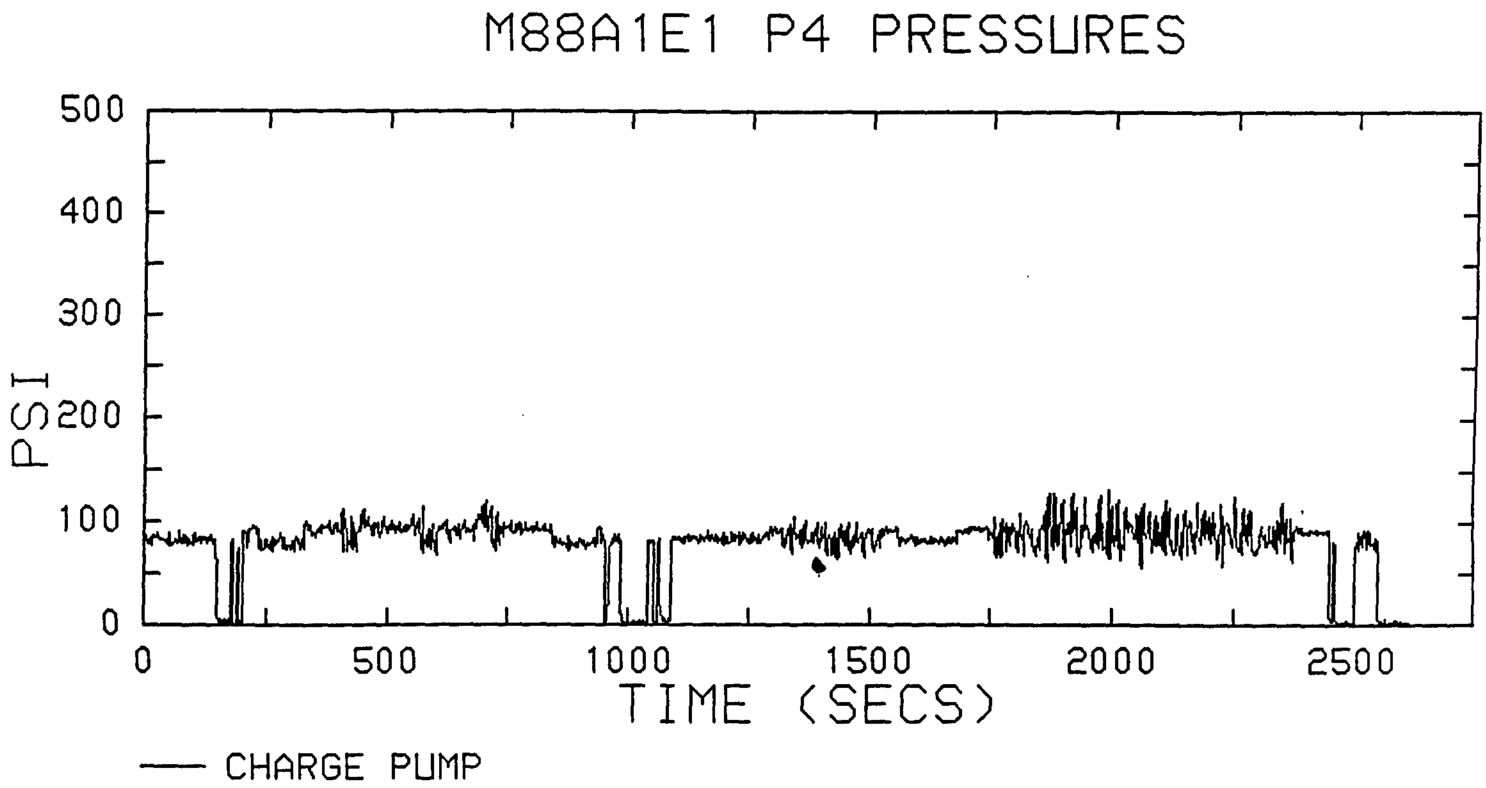

M88A1E1 P4 PRESSURES
PSI
5000
4000
3000
2000
1000
0
0
500
1000
1500
2000
2500
TIME (SECS)
PUMP COMPENSATION



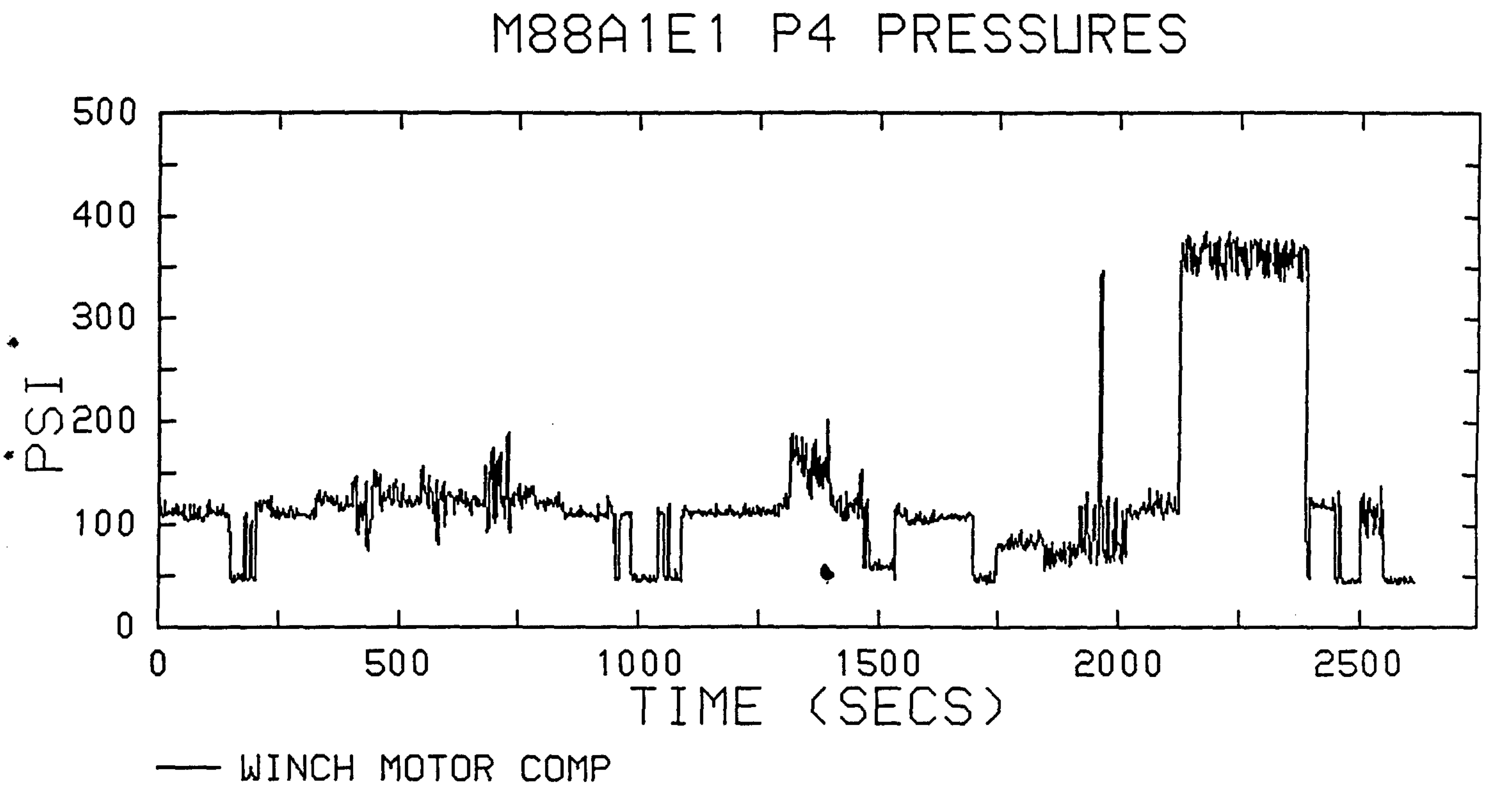
M88A1E1 P4 PRESSURES
PSI
5000
4000
3000
2000
1000
0
0
500
1000
1500
2000
2500
TIME (SECS)
MAIN OUT

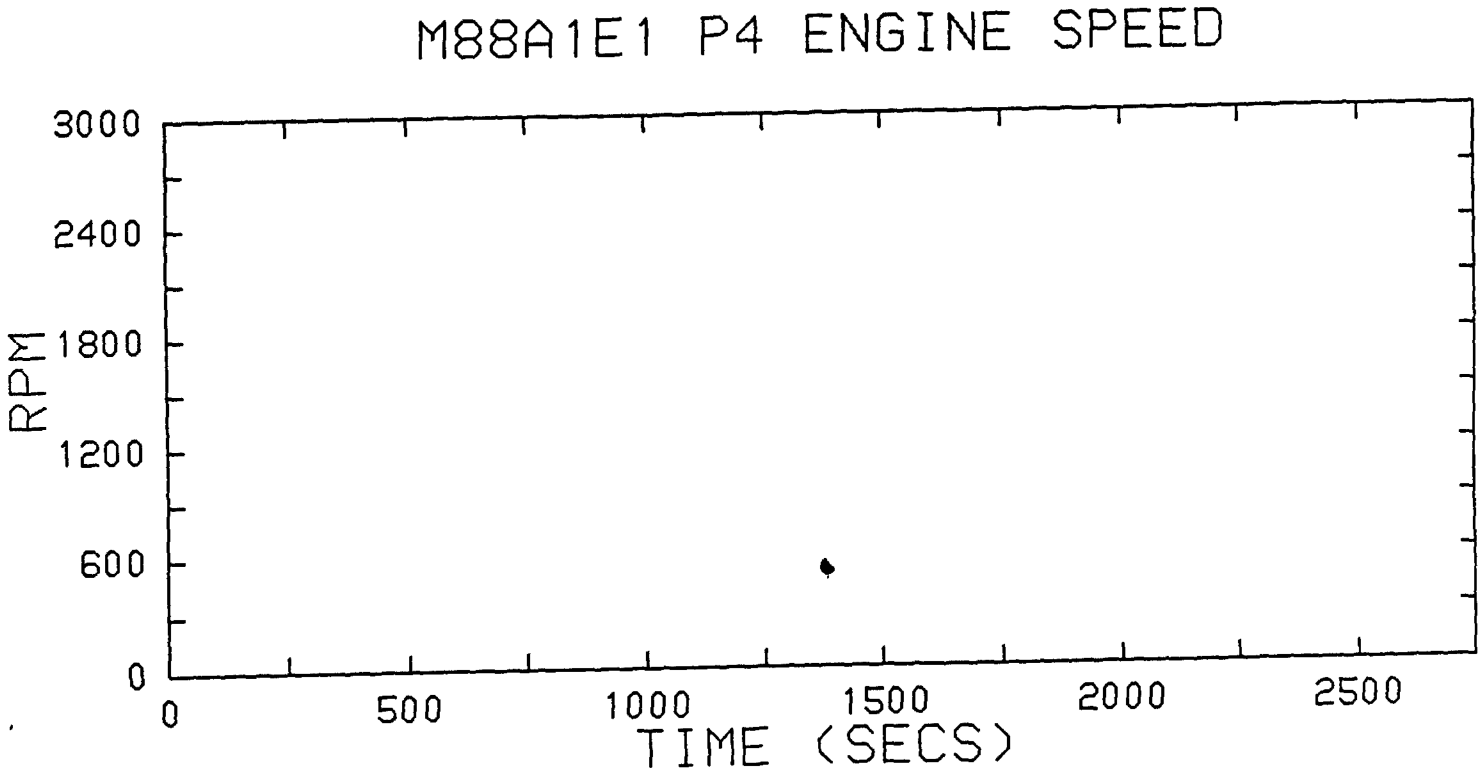


M88A1E1 P4 PRESSURES
5000
4000
3000
2000
1000
0
PSI
0
500
1000
1500
2000
2500
TIME (SECS)
MAIN IN



M88A1E1 P4 PRESSURES
PSI
500
400
300
200
100
0
0
500
1000
1500
2000
2500
TIME (SECS)
CHARGE PUMP

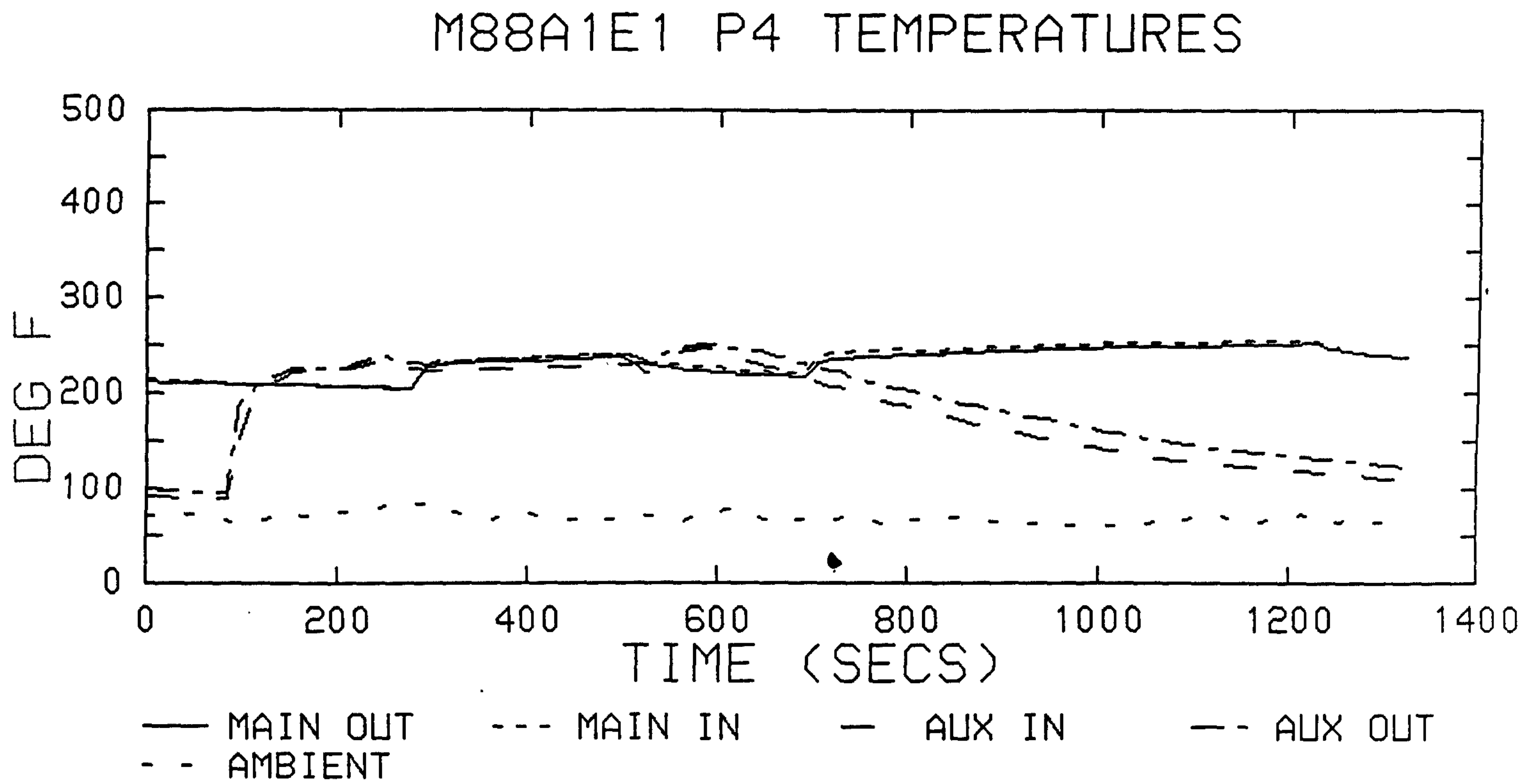

M88A1E1 P4 PRESSURES
PSI
0
100
200
300
400
500
0
500
1000
1500
2000
2500
TIME (SECS)
WINCH MOTOR COMP


M88A1E1 P4 ENGINE SPEED
RPM
3000
2400
1800
1200
600
0
0
500
1000
1500
2000
2500
TIME (SECS)

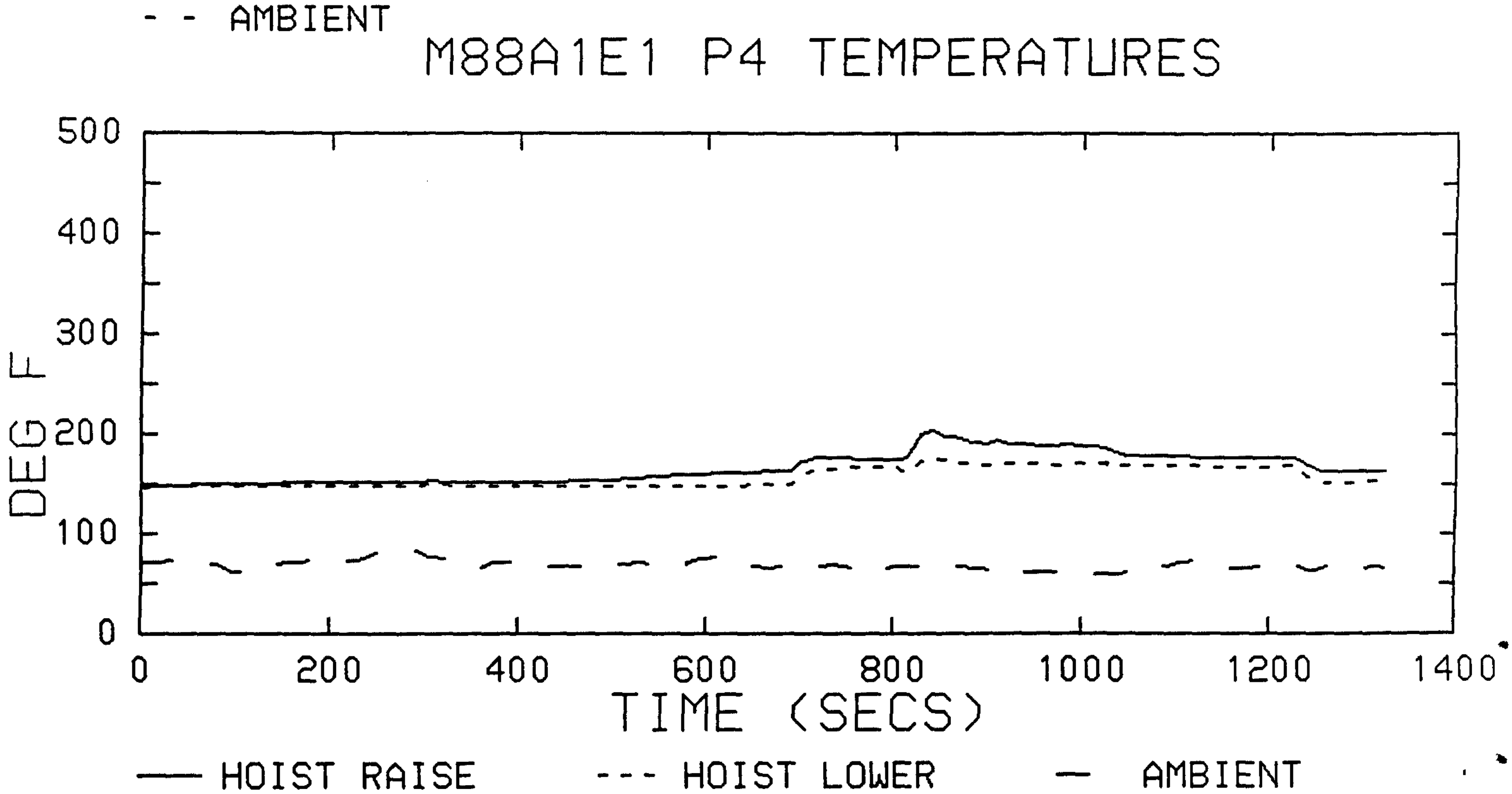
TRIAL #81

M88A1E1 (vehicle P4)

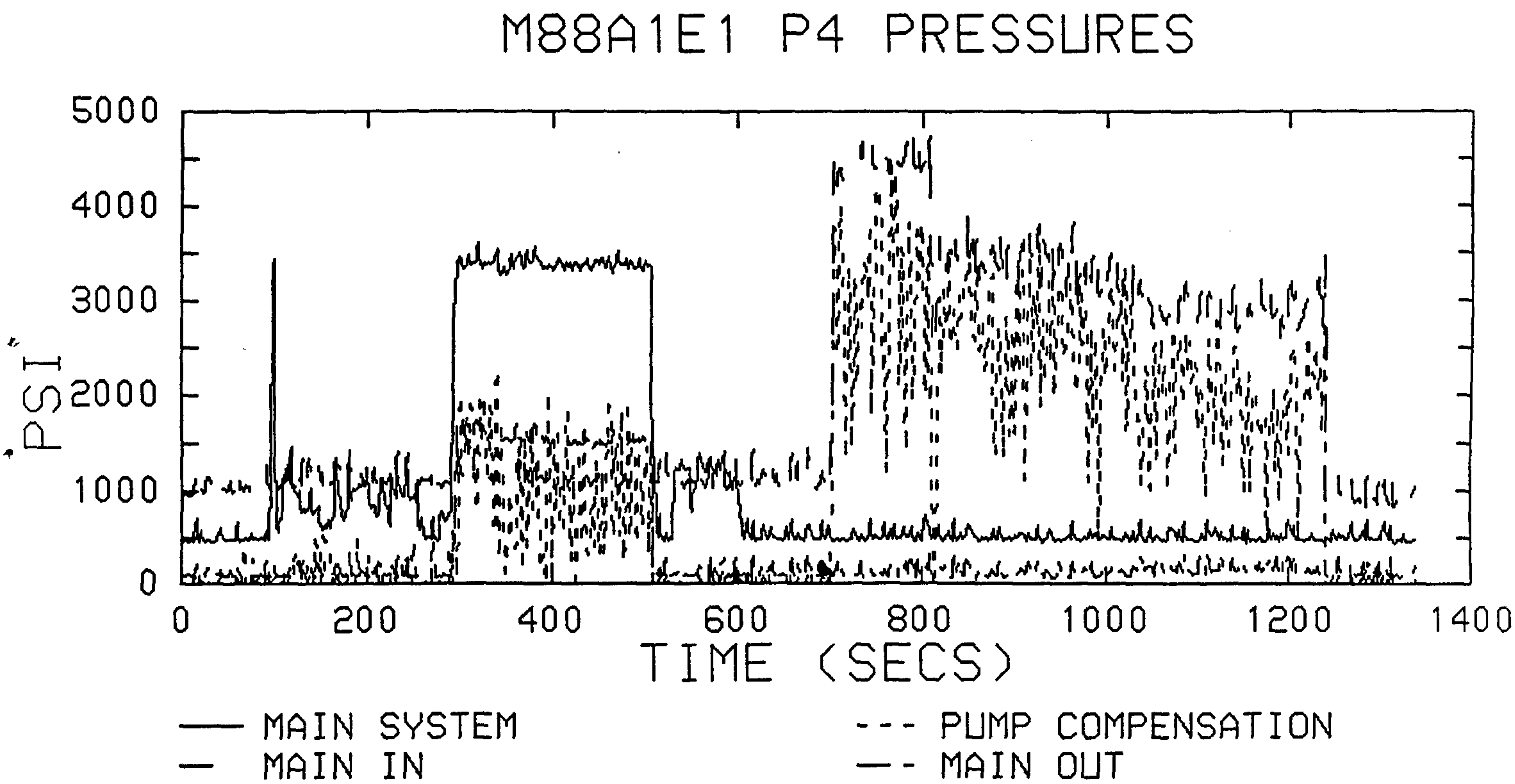
29 March 91

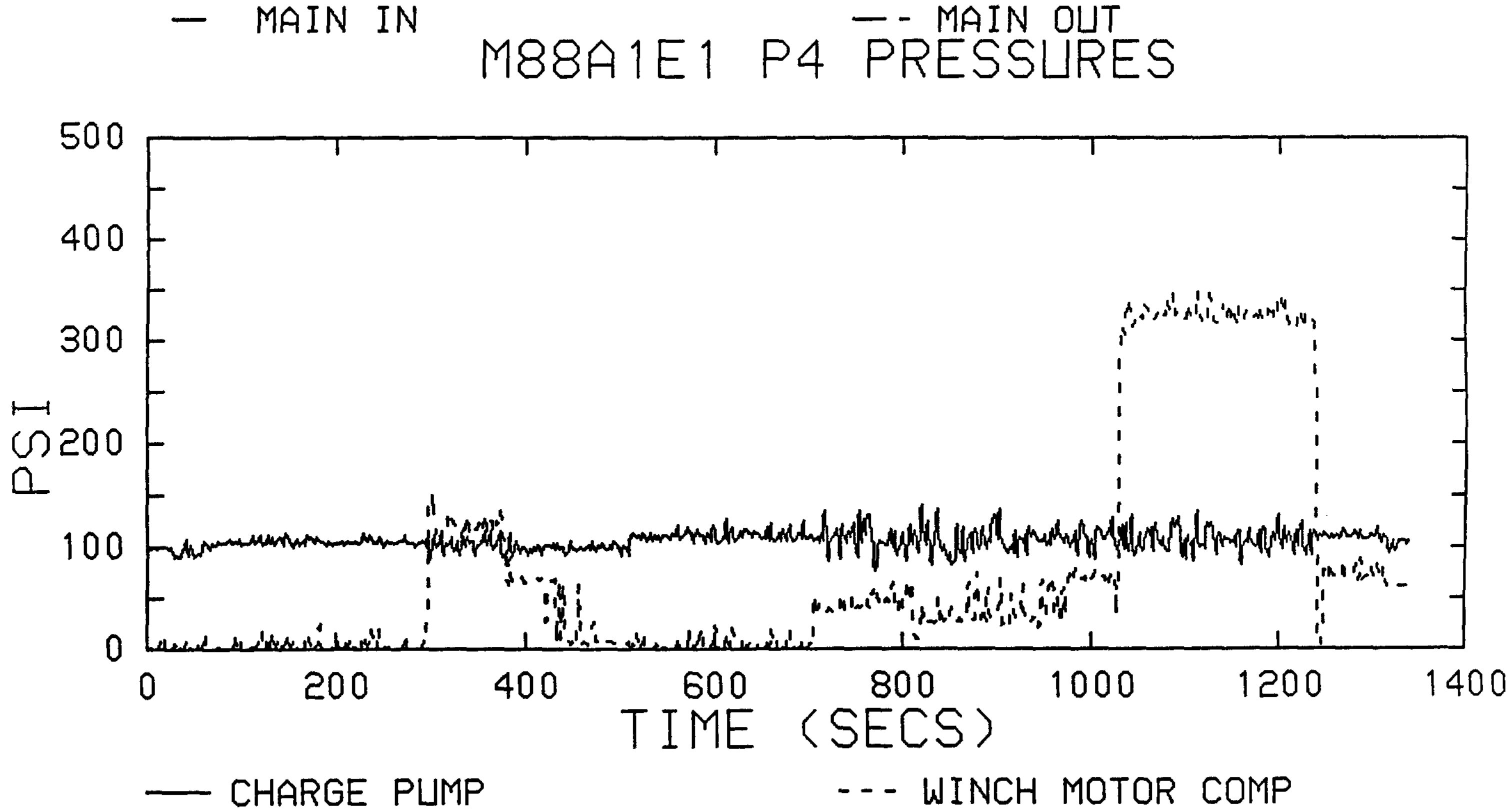


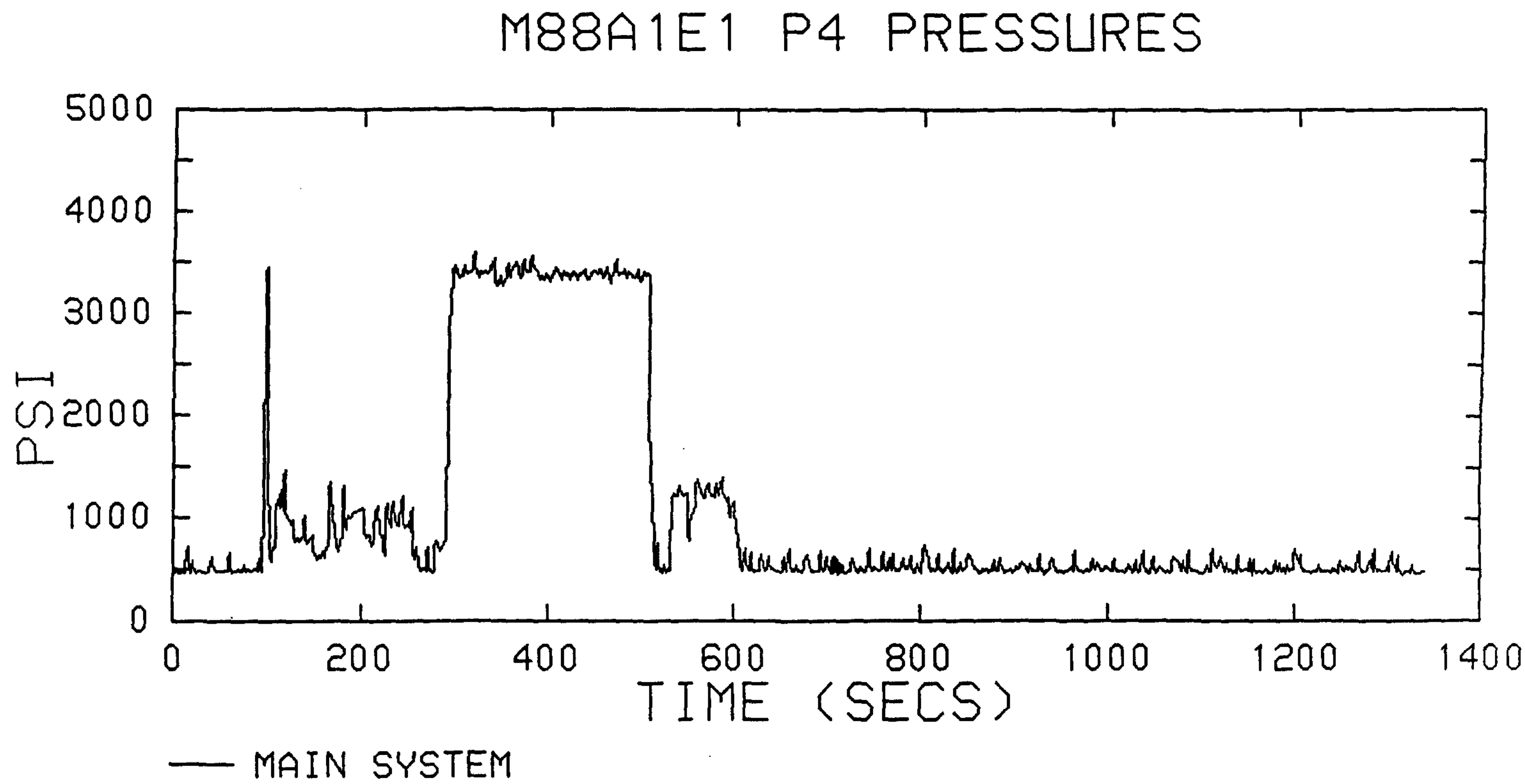
M88A1E1 P4 TEMPERATURES
500
400
300
200
100
0
DEG F
0
200
400
600
800
1000
1200
1400
TIME (SECS)
MAIN OUT
MAIN IN
AUX IN
AUX OUT
AMBIENT

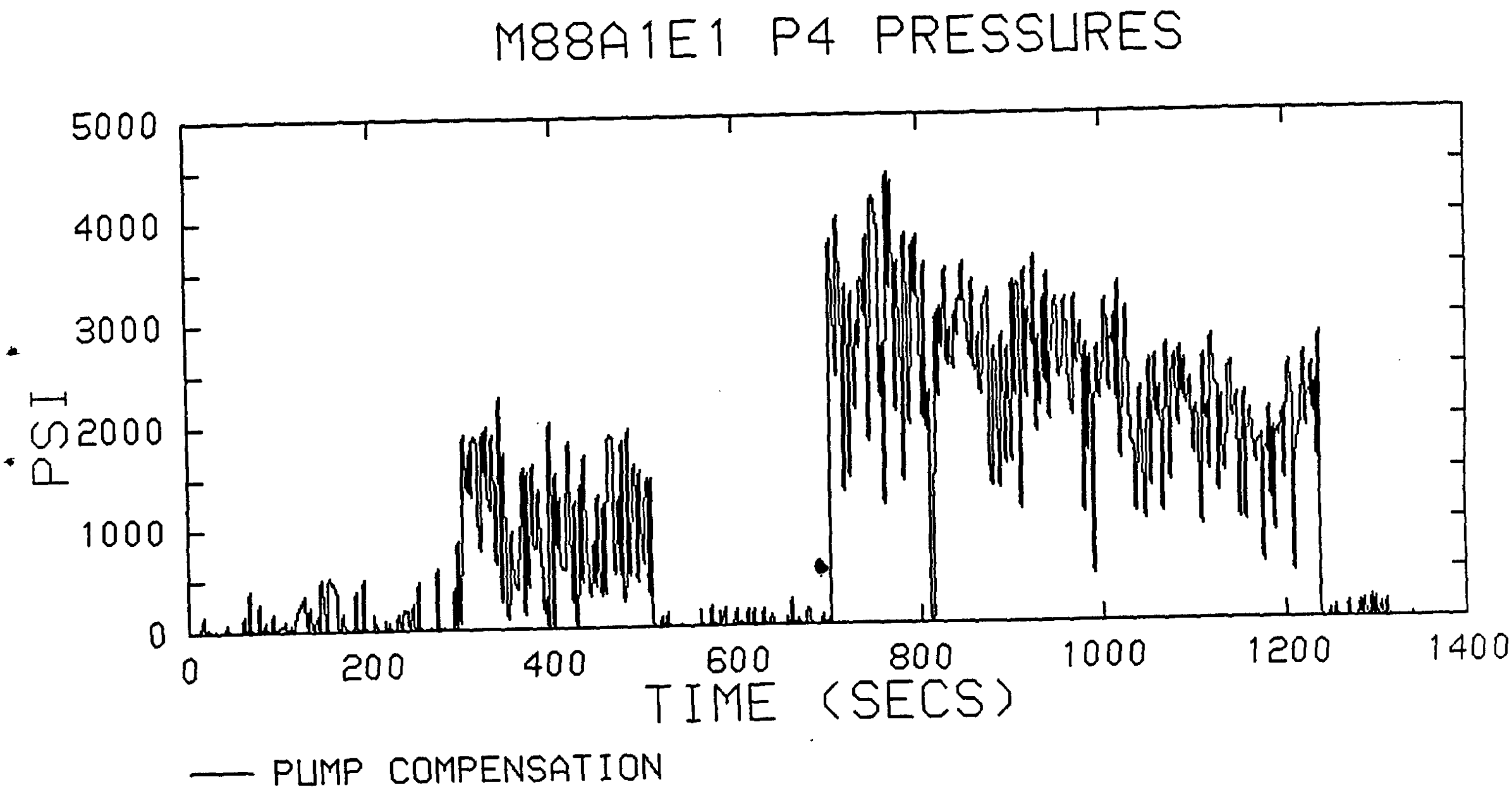


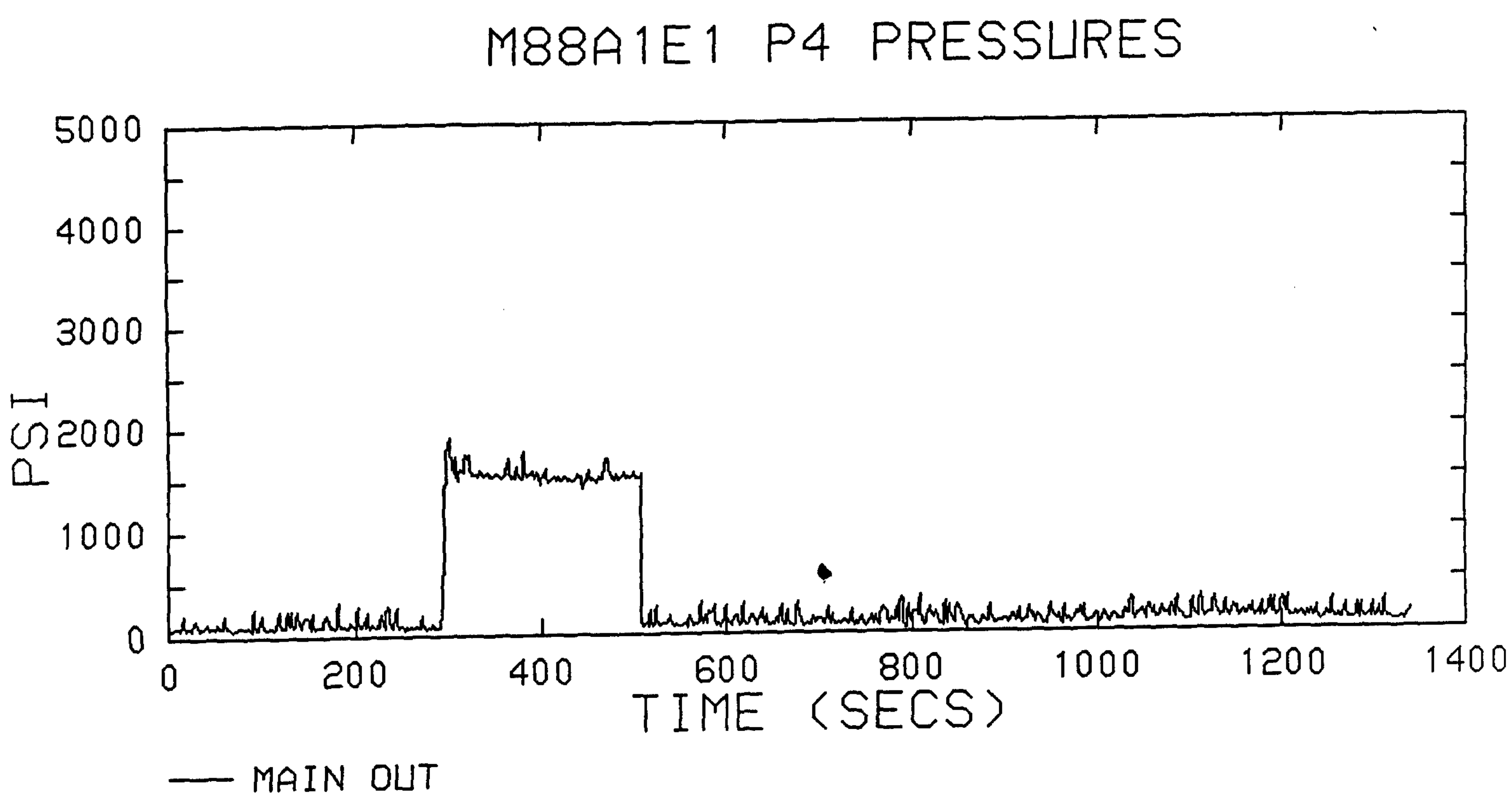
M88A1E1 P4 TEMPERATURES
500
400
300
200
100
0
DEG F
0
200
400
600
800
1000
1200
1400
TIME (SECS)
HOIST RAISE
HOIST LOWER
AMBIENT

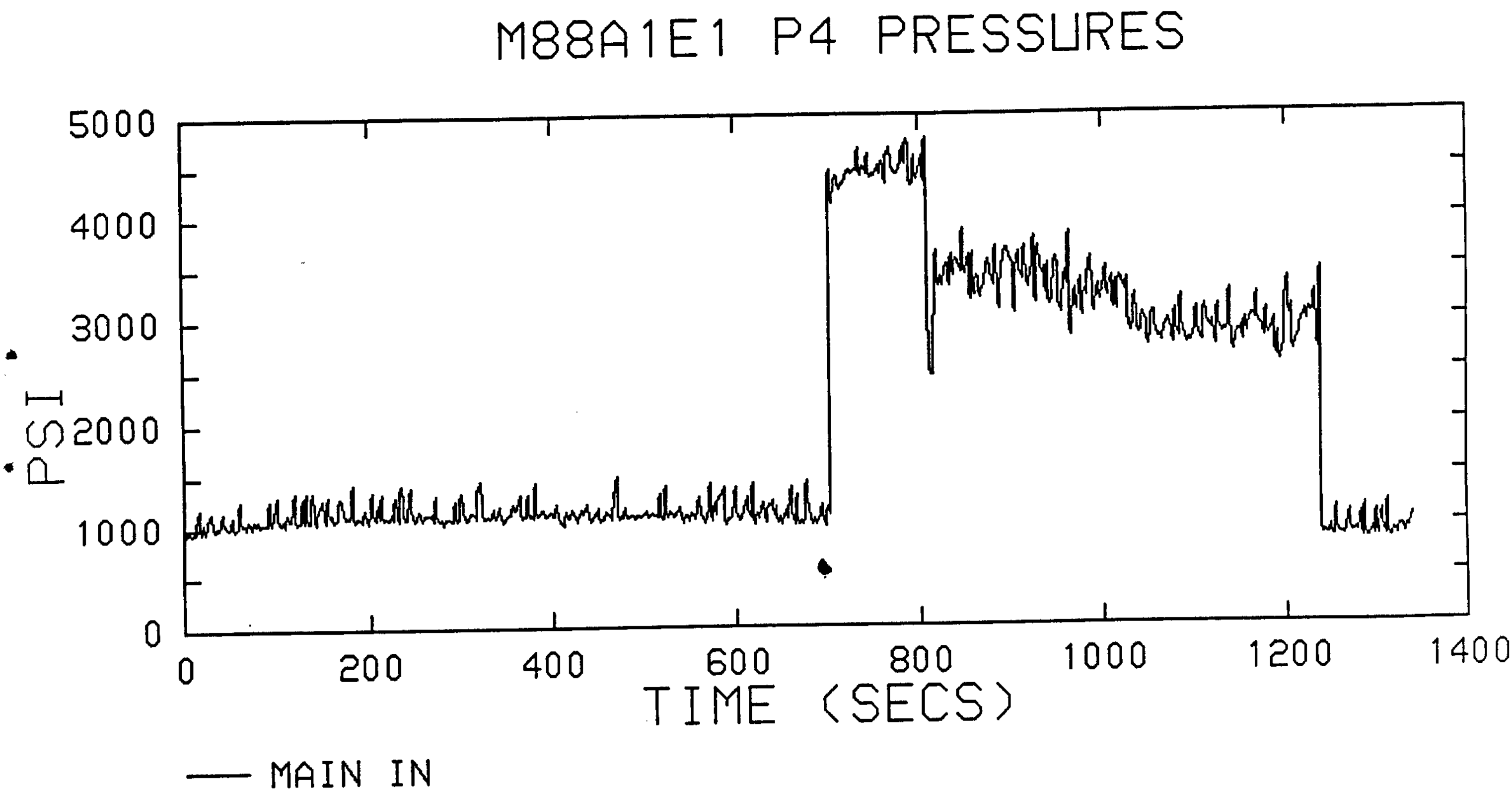

M88A1E1 P4 PRESSURES
5000
4000
3000
2000
1000
0
PSI
0
200
400
600
800
1000
1200
1400
TIME (SECS)
MAIN SYSTEM
MAIN IN
PUMP COMPENSATION
MAIN OUT

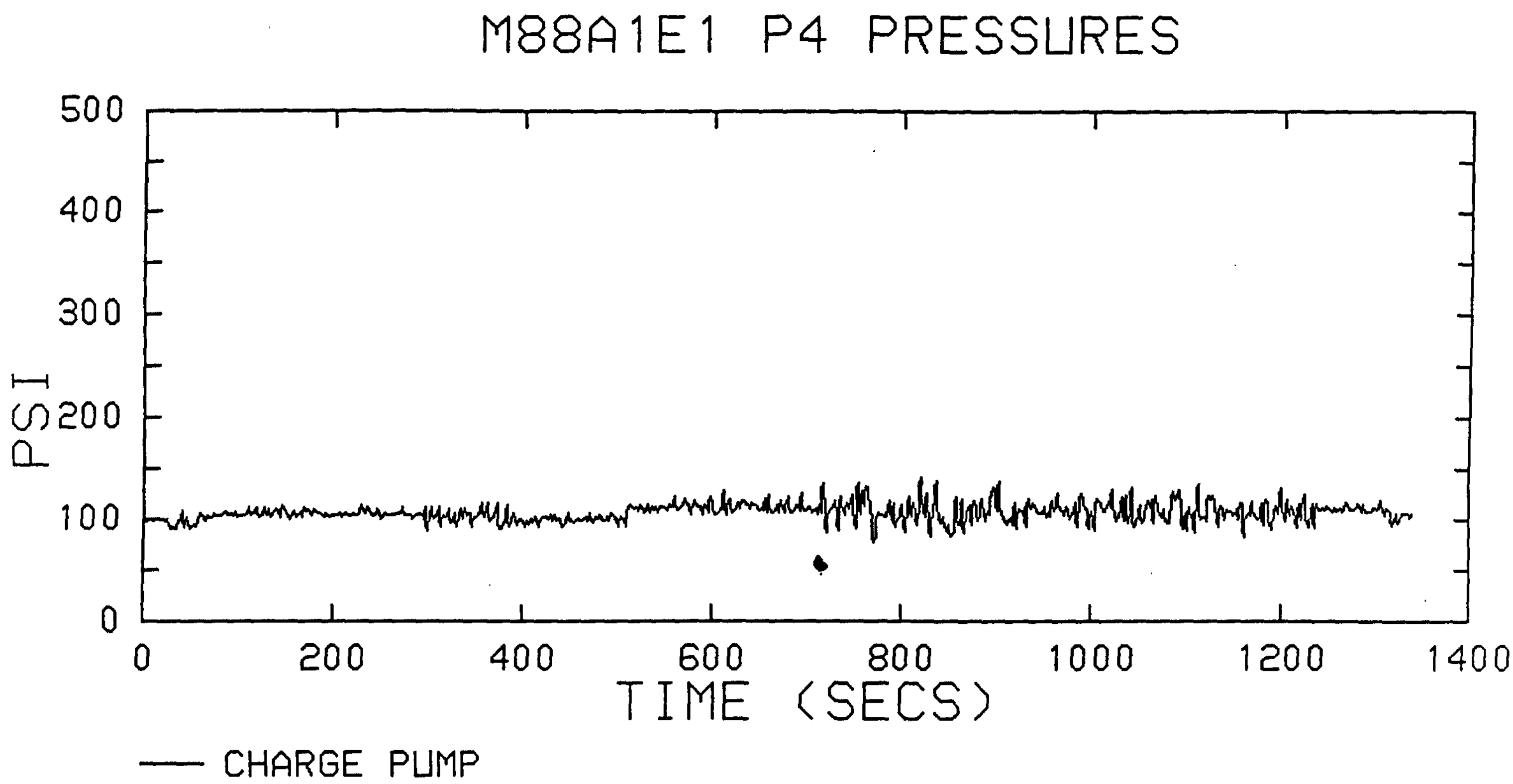

M88A1E1 P4 PRESSURES
500
400
300
200
100
0
PSI
0
200
400
600
800
1000
1200
1400
TIME (SECS)
CHARGE PUMP
WINCH MOTOR COMP

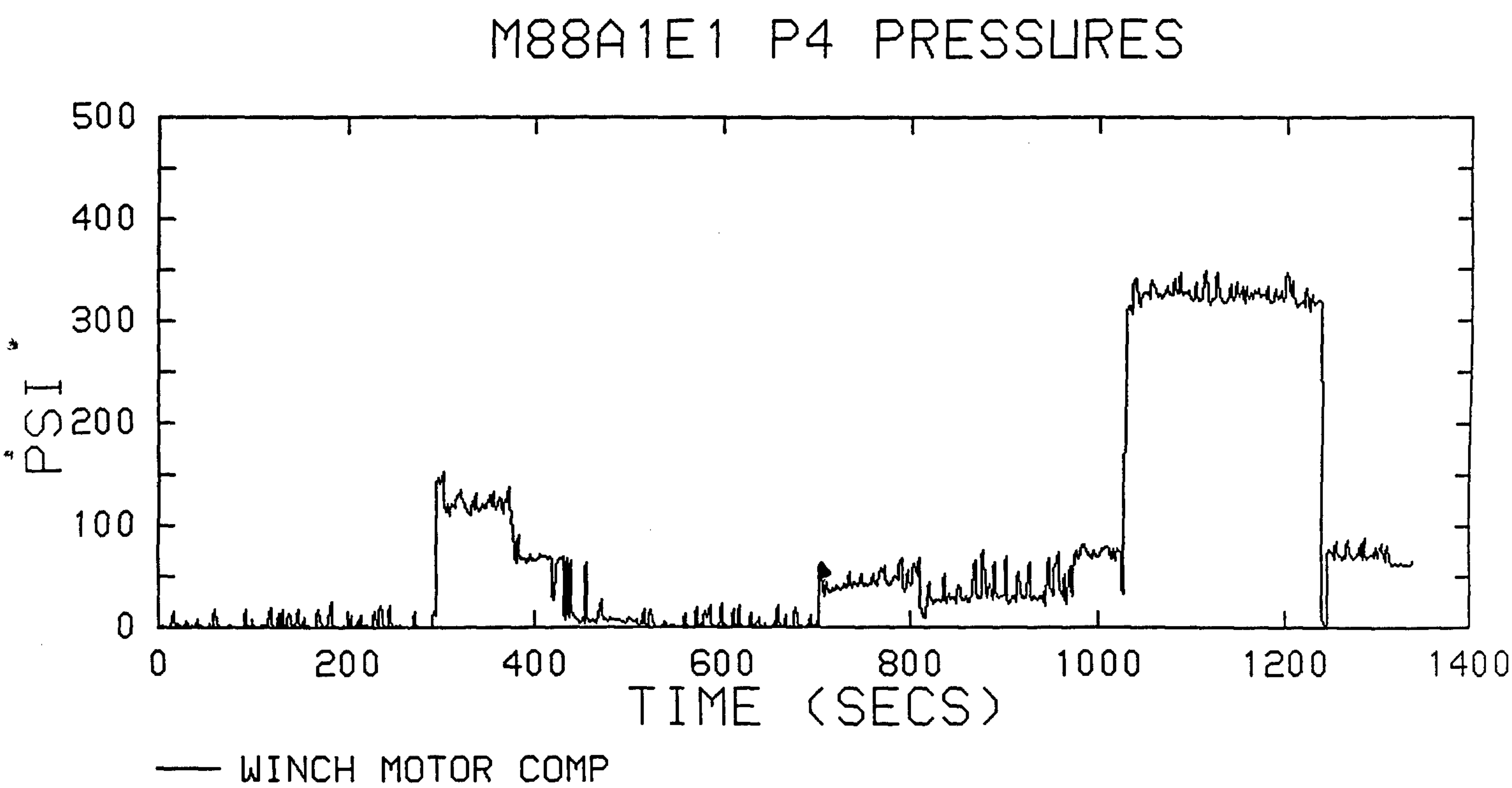

M88A1E1 P4 PRESSURES
PSI
5000
4000
3000
2000
1000
0
0
200
400
600
800
1000
1200
1400
TIME (SECS)
MAIN SYSTEM

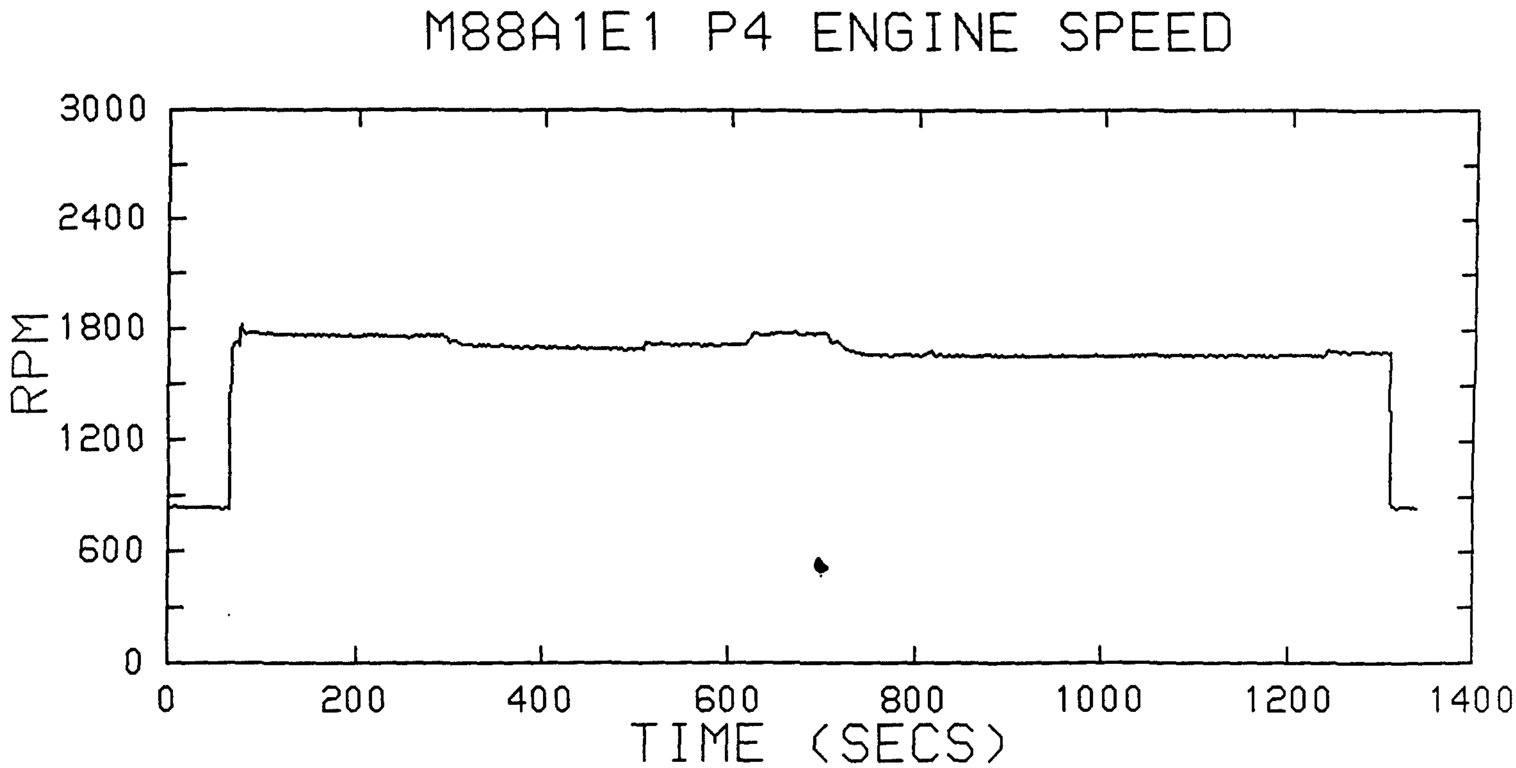

M88A1E1 P4 PRESSURES
PSI
5000
4000
3000
2000
1000
0
0
200
400
600
800
1000
1200
1400
TIME (SECS)
PUMP COMPENSATION

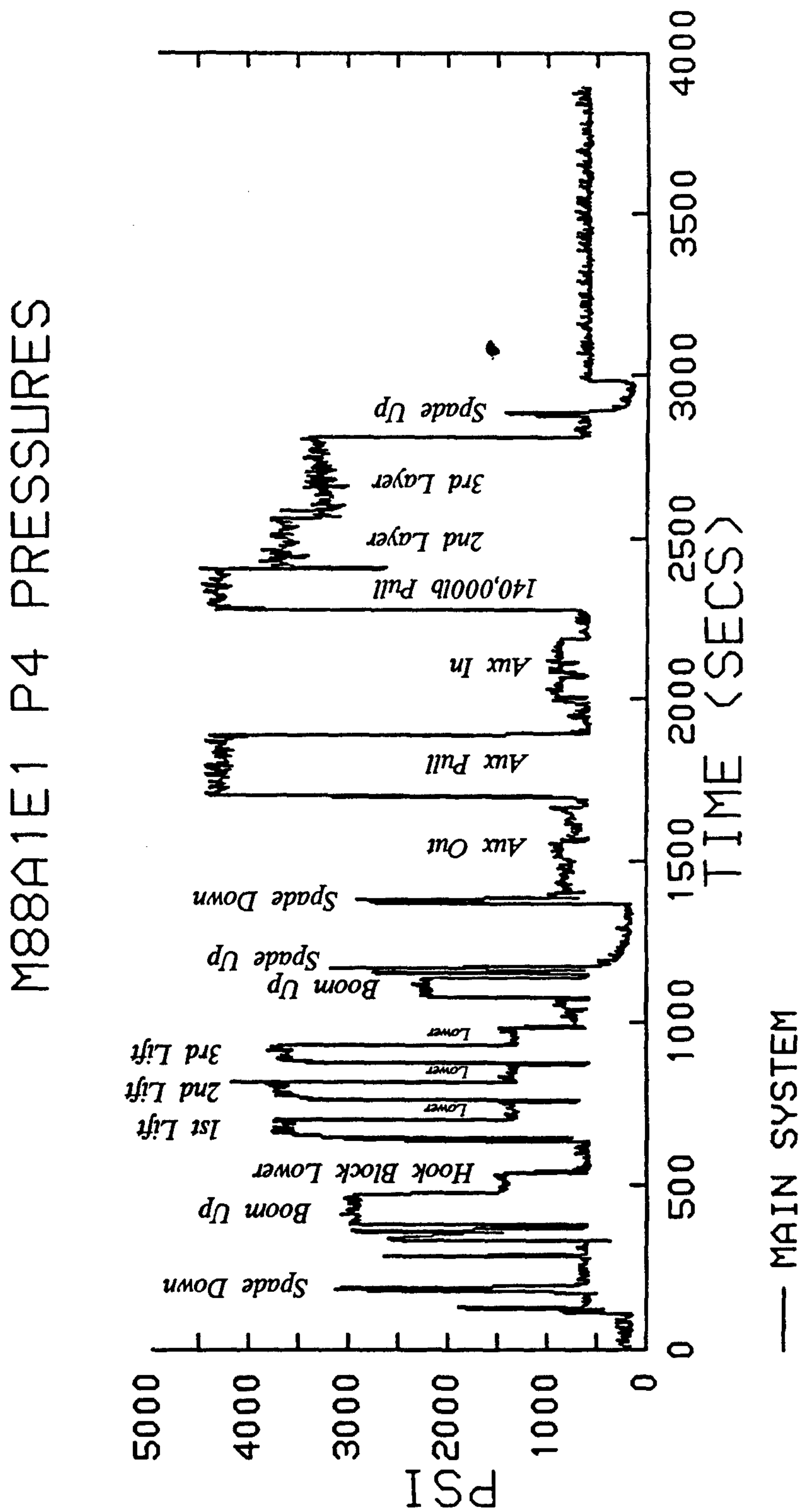

M88A1E1 P4 PRESSURES
5000
4000
3000
2000
1000
0
PSI
0
200
400
600
800
1000
1200
1400
TIME (SECS)
MAIN OUT


M88A1E1 P4 PRESSURES
5000
4000
3000
2000
1000
0
PSI
0
200
400
600
800
1000
1200
1400
TIME (SECS)
MAIN IN


M88A1E1 P4 PRESSURES
500
400
300
200
100
0
PSI
0
200
400
600
800
1000
1200
1400
TIME (SECS)
CHARGE PUMP


M88A1E1 P4 PRESSURES
PSI
500
400
300
200
100
0
0
200
400
600
800
1000
1200
1400
TIME (SECS)
WINCH MOTOR COMP


M88A1E1 P4 ENGINE SPEED
RPM
3000
2400
1800
1200
600
0
0
200
400
600
800
1000
1200
1400
TIME (SECS)



M88A1E1 P4 PRESSURES
PSI
5000
4000
3000
2000
1000
0
0
500
1000
1500
2000
2500
3000
3500
4000
TIME (SECS)
Spade Down
Boom Up
Hook Block Lower
1st Lift
2nd Lift
3rd Lift
Lower
Lower
Lower
Boom Up
Spade Up
Spade Down
Aux Out
Aux Pull
Aux In
140,000lb Pull
2nd Layer
3rd Layer
Spade Up
MAIN SYSTEM

FIGURE 1

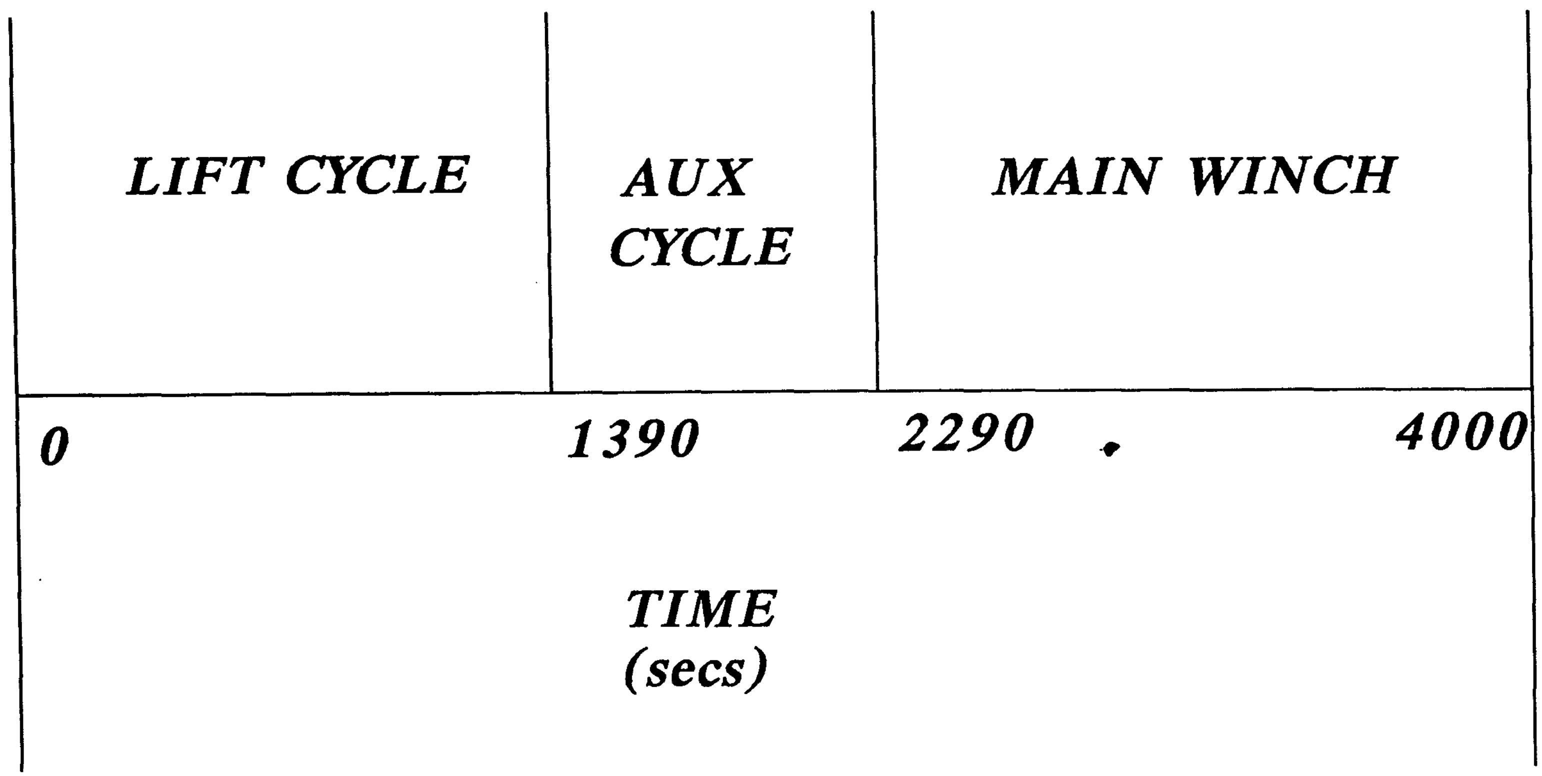


FIGURE 2

APPENDIX I

Three-Pump Modification

APPENDIX I

The T.H. Paris Corp, BMY, and TACOM worked closely in developing a three-pump system for the purpose of reducing the heat gain in the system hydraulic oil during vehicle hydraulic operations. T.H. Paris supplied one three-pump system which was installed and tested in vehicle P4. T.H. Paris also developed system drawings and parts lists for the new design. The new three-pump drawings and parts lists include:

T.H. Paris Drawing #	Drawing Title
B 957	Hydraulic Circuit Modification: Hose Assembly Identification
B 958	Hose Circuit modification: Hose Specifications
B 959	Hydraulic Circuit Modification: Fitting Identification
B 960	Hydraulic Circuit Modification: Fitting Specifications
B 961	Hydraulic Circuit: Modified Configuration
B 962	Hydraulic Circuit: Original Configuration

Additional three-pump data includes:

1. Bill of Materials (for complete modification).

2. Rexroth drawing SK43-A53-0402-E-2, "Outline Drawing M88A1E1 Triple Pump."

3. Rexroth drawing SK-43-A53-0406-B-0, "Welded Pipe Assembly for M88A1E1 Gear Pump."

4. Rexroth drawing SK43-A53-0407-B-1, "Welded Pipe Assembly M88A1E1 A10V Pump."

5. Rexroth drawing SK43-A53-0408-C-1, "M88A1E1 Suction Header Block."

6. Rexroth drawing SK43-A53-0409-C-0, "Triple Pump Support Bracket M88A1E1 Tank Recovery Vehicle."

Test data was analyzed to determine the benefit gained by installing the three-pump system. During the aux winch cycle, the following observation was made.

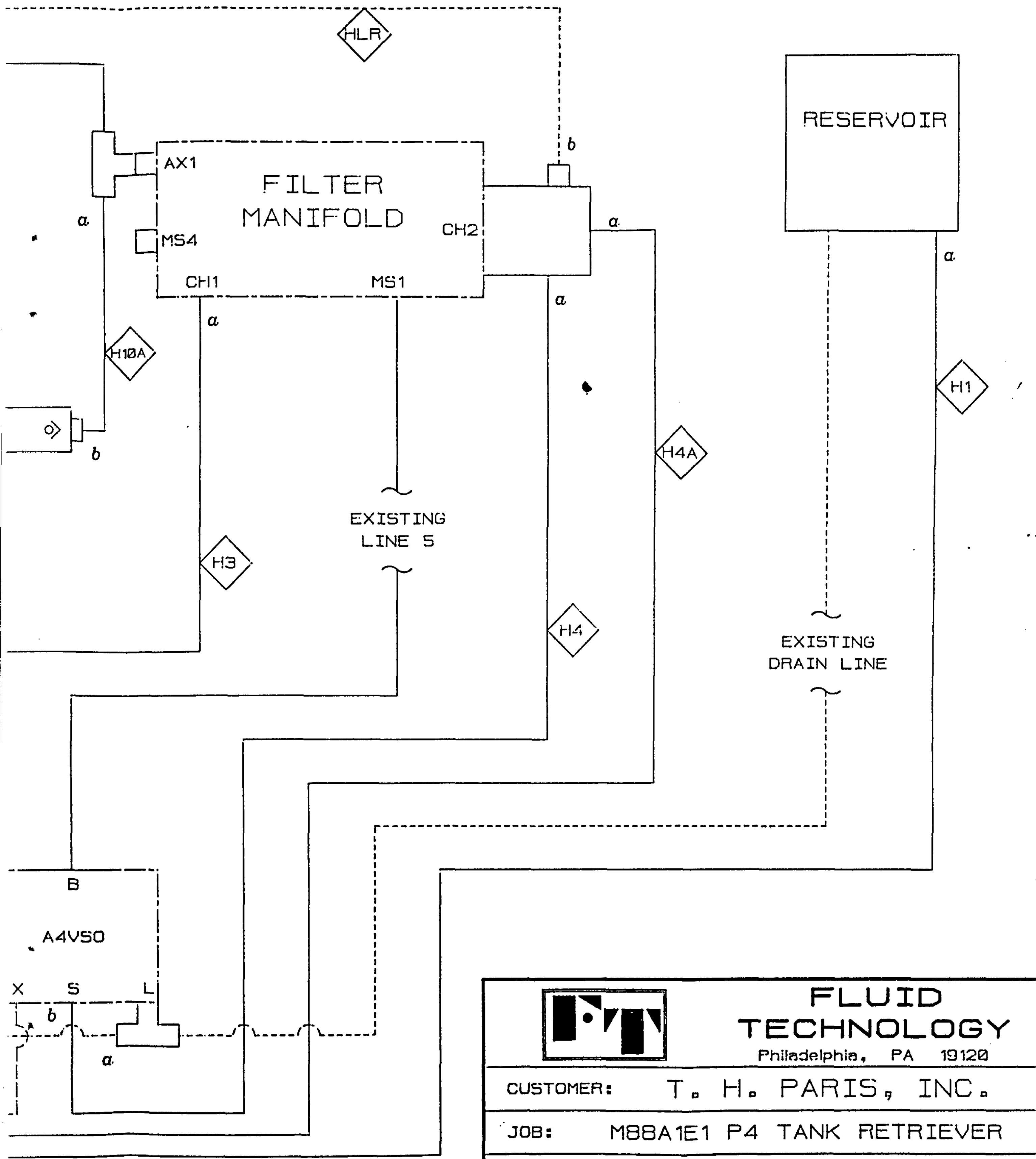
Run #	Aux Cycle Time (min)	Temp Rise °F
64	12	28
65	17	7
66	13	24
67	19	15
68	9	14
69	14	37
70	12	22
71	10	20

Average Gain = 1.73 °F/min

Run #	Aux Cycle Time (min)	Temp Rise °F
74	18	23
75	11	16
76	21	22
77	11	24
78	11	-5
79	11	13

Average Gain = 1.12 °F/min

Comparison of the data indicates a 35% reduction in heat gain during the aux winch cycle is realized by installing a three-pump verses a two-pump system.



CATION
ORMATION

FLUID TECHNOLOGY
Philadelphia, PA 19120

CUSTOMER: T. H. PARIS, INC.

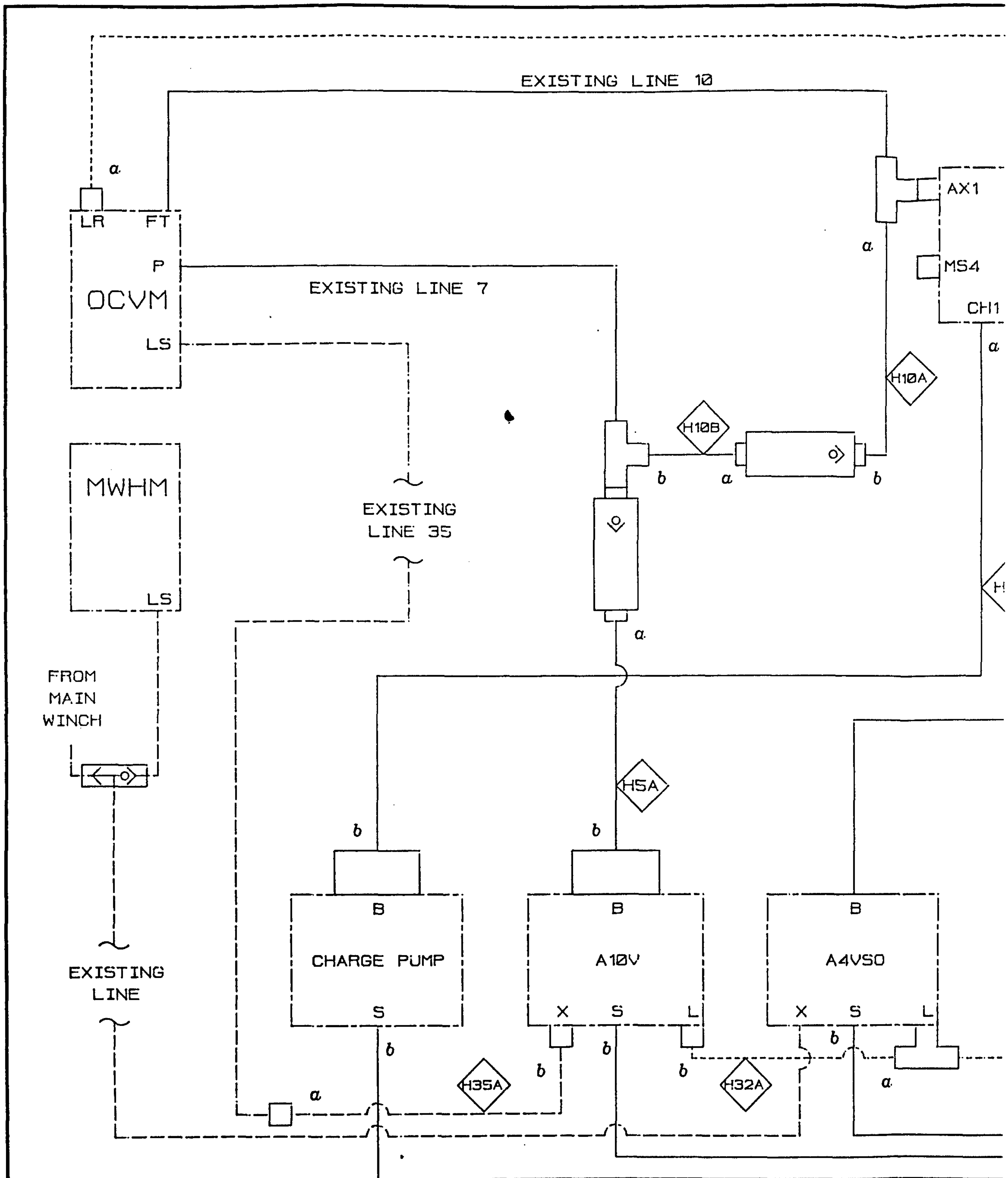
JOB: M88A1E1 P4 TANK RETRIEVER

TITLE: HYDRAULIC CIRCUIT MODIFICATION
HOSE ASSEMBLY IDENTIFICATION

DWG: B 957

DATE: 04/10/91

BY: DP



NOTES: REFER TO DWG. B 958 FOR HOSE ASSEMBLY SPECIFICATION
REFER TO DWGS. B 959 & B 960 FOR FITTINGS INFORMATION
"a" & "b" INDICATE HOSE ASSEMBLY ORIENTATION

HOSE END "6"	HOSE TYPE	AEROQUIP HOSE
SPLIT FLG. - CODE 61 STR.; -24	SUCTION	300 SERIES
SPLIT FLG. - CODE 61 90 DEG. EL; -20	HIGH PSI	FC254
SPLIT FLG. - CODE 61 90 DEG. EL; -12	HIGH PSI	2781
SPLIT FLG. - CODE 61 45 DEG. EL; -24	SUCTION	300 SERIES
SPLIT FLG. - CODE 61 STR.; -24	SUCTION	300 SERIES
SAE 37 DEG. FEM. SWIV. 90 DEG. EL LG; -8	MED PSI	
SAE 37 DEG. FEM. SWIV. 90 DEG. EL; -4	HIGH PSI	2781
SAE 37 DEG. FEM. SWIV. 45 DEG. EL; -12	HIGH PSI	2781
SAE 37 DEG. FEM. SWIV. 45 DEG. EL; -12	HIGH PSI	2781
SAE 37 DEG. FEM. SWIV. 90 DEG. EL; -12	MED PSI	

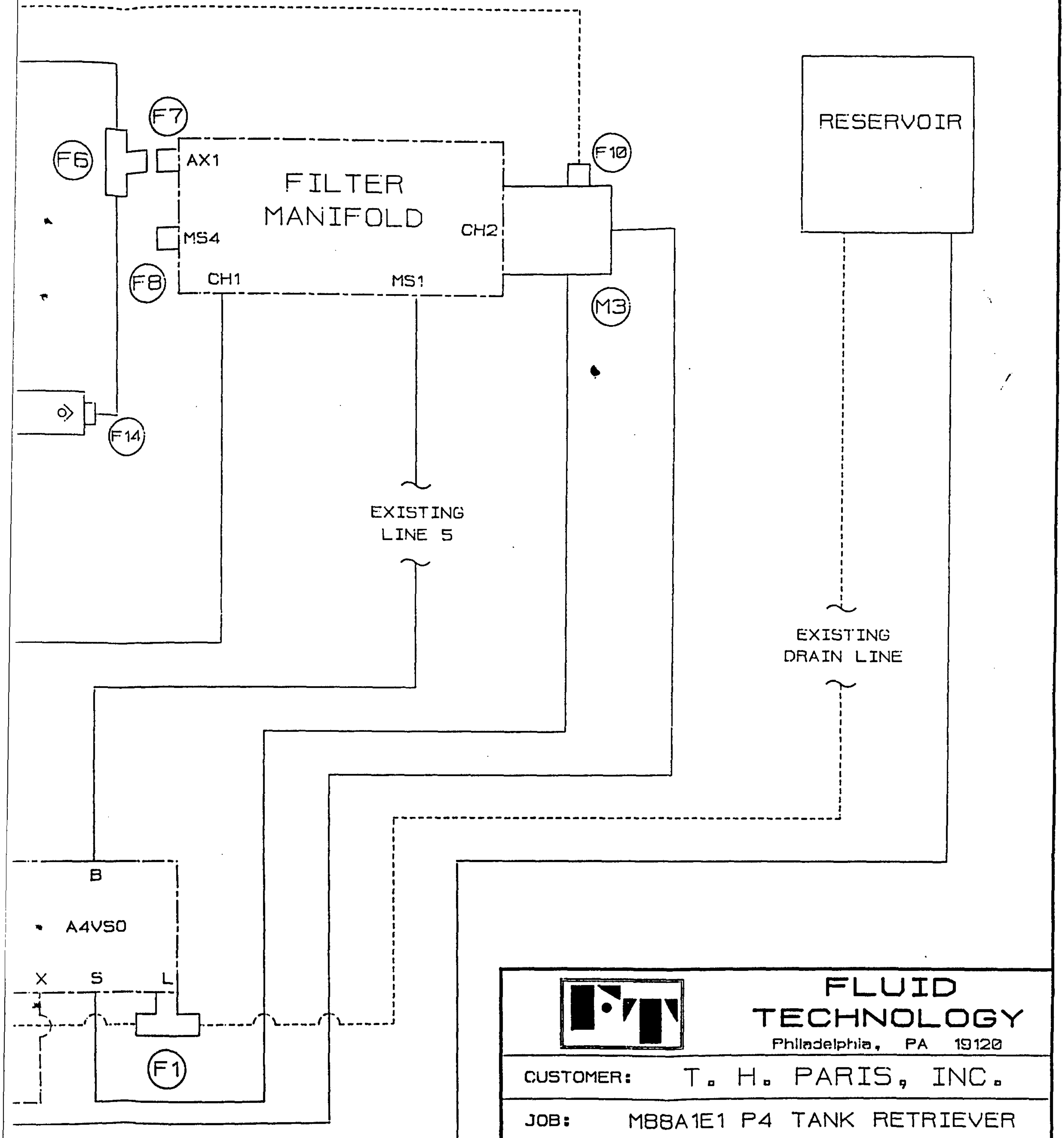
EMBLY LOCATION
FITTINGS INFORMATION
ORIENTATION

FLUID TECHNOLOGY
Philadelphia, PA 19120

CUSTOMER:	T. H. PARIS, INC.
JOB:	M88A1E1 P4 TANK RETRIEVER
TITLE:	HYDRAULIC CIRCUIT MODIFICATION HOSE SPECIFICATIONS
DWG: B 958	DATE: 04/06/91 BY: DP

HOSE ASSEMBLY	HOSE I.D. (in.)	CUT HOSE LENGTH (in.)	HOSE END "a"	
H1	1.50"	25.0"	SPLIT FLG. – CODE 61 45 DEG. EL; -24	SPLIT FLG. STR.; -24
H3	1.00"	30.0"	SPLIT FLG. – CODE 61 STR.; -16	SPLIT FLG. 90 DEG. EL;
H5A	0.75"	40.0"	SAE 37 DEG. FEM. SWIV. STR.; -12	SPLIT FLG. 90 DEG. EL;
H4	1.50"	15.5"	SPLIT FLG. – CODE 61 STR.; -24	SPLIT FLG. 45 DEG. EL;
H4A	1.25"	23.0"	SPLIT FLG. – CODE 61 90 DEG. EL; -20	SPLIT FLG. STR.; -24
H32A	0.50"	27.0"	SAE 37 DEG. FEM. SWIV. STR.; -16	SAE 37 DEG. 90 DEG. EL
H35A	0.25"	40.0"	SAE 37 DEG. FEM. SWIV. STR.; -4	SAE 37 DEG. 90 DEG. EL;
H10A	0.75"	19.0"	SAE 37 DEG. FEM. SWIV. STR.; -12	SAE 37 DEG. 45 DEG. EL;
H10B	0.75"	24.0"	SAE 37 DEG. FEM. SWIV. STR.; -12	SAE 37 DEG. 45 DEG. EL;
HLR	0.75"	180.0"	SAE 37 DEG. FEM. SWIV. STR.; -12	SAE 37 DEG. 90 DEG. EL;

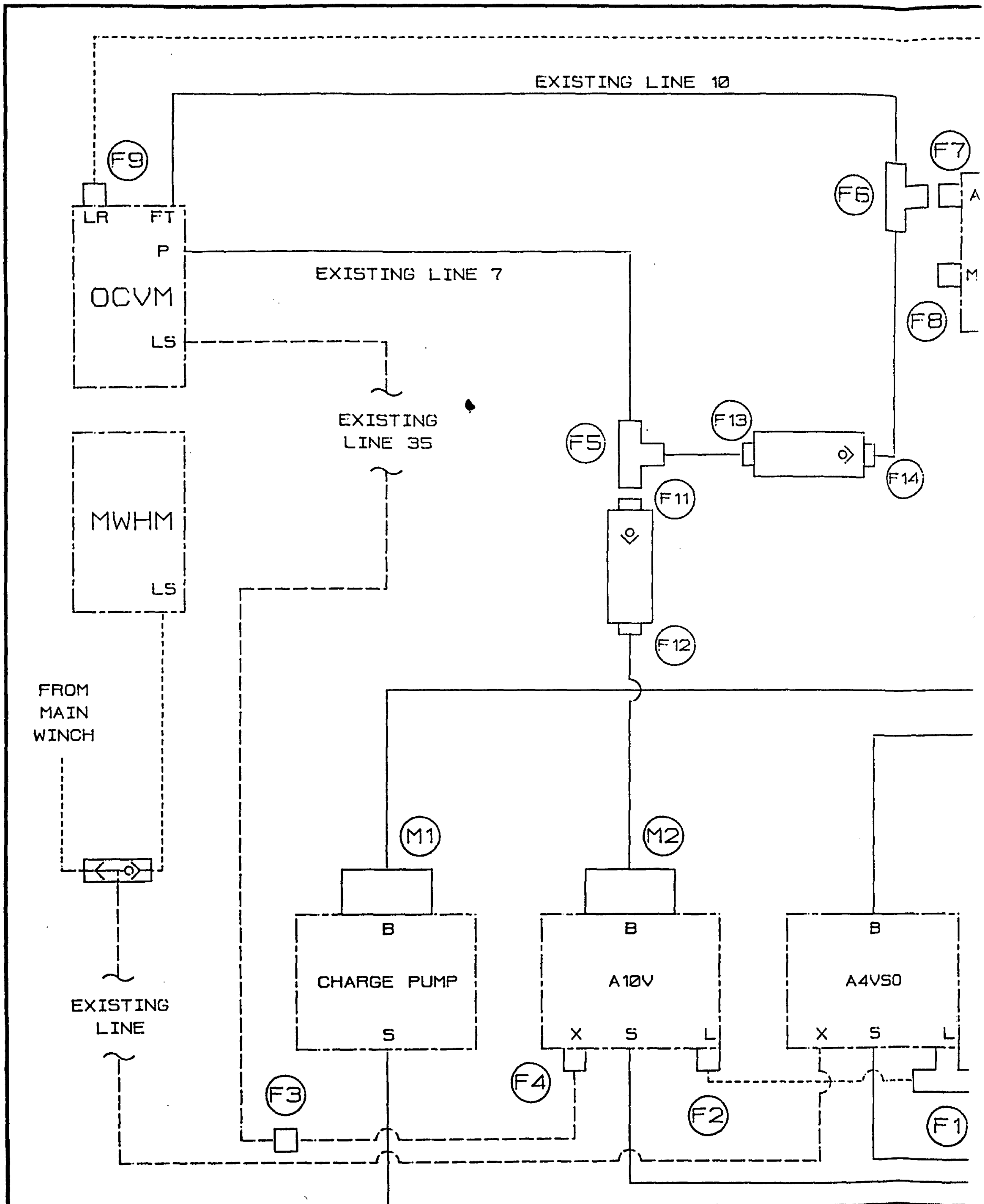
NOTES: REFER TO DWG. B 957 FOR HOSE ASSEMBLY LOCATION
REFER TO DWGS. B 959 & B 960 FOR FITTINGS INFORM
"a" & "b" INDICATE HOSE ASSEMBLY ORIENTATION



NS
MBLY INFORMATION

FLUID TECHNOLOGY
Philadelphia, PA 19120

CUSTOMER:	T. H. PARIS, INC.
JOB:	M88A1E1 P4 TANK RETRIEVER
TITLE:	HYDRAULIC CIRCUIT MODIFICATION FITTING IDENTIFICATION
DWG:	B 959
DATE:	04/10/91
BY:	DP



NOTES: REFER TO DWG. B 960 FOR FITTINGS SPECIFICATIONS
REFER TO DWGS. B 957 & B 958 FOR HOSE ASSEMBLY INFORMATI(

THREAD TYPE & SIZE (DASH NUMBER)	
SAE "O" RING MALE -16	2 X (SAE 37 DEG. MALE -16)
SAE "O" RING MALE -10	SAE 37 DEG. MALE -8
SAE 37 DEG. MALE -4	SAE 37 DEG. MALE -4
SAE "O" RING MALE -4	SAE 37 DEG. MALE -4
SAE 37 DEG. FEM. SWIV. -12	2 X (SAE 37 DEG. MALE -12)
SAE 37 DEG. FEM. SWIV. -12	2 X (SAE 37 DEG. MALE -16)
SAE "O" RING MALE -12	SAE 37 DEG. MALE -12
SAE "O" RING MALE -12	
SAE "O" RING MALE -12	SAE 37 DEG. MALE -12
SAE "O" RING MALE -12	SAE 37 DEG. MALE -12
SAE "O" RING MALE -12	SAE 37 DEG. MALE -12
SAE "O" RING MALE -12	SAE 37 DEG. MALE -12
SAE "O" RING MALE -12	SAE 37 DEG. MALE -12
SAE "O" RING MALE -12	SAE 37 DEG. MALE -12

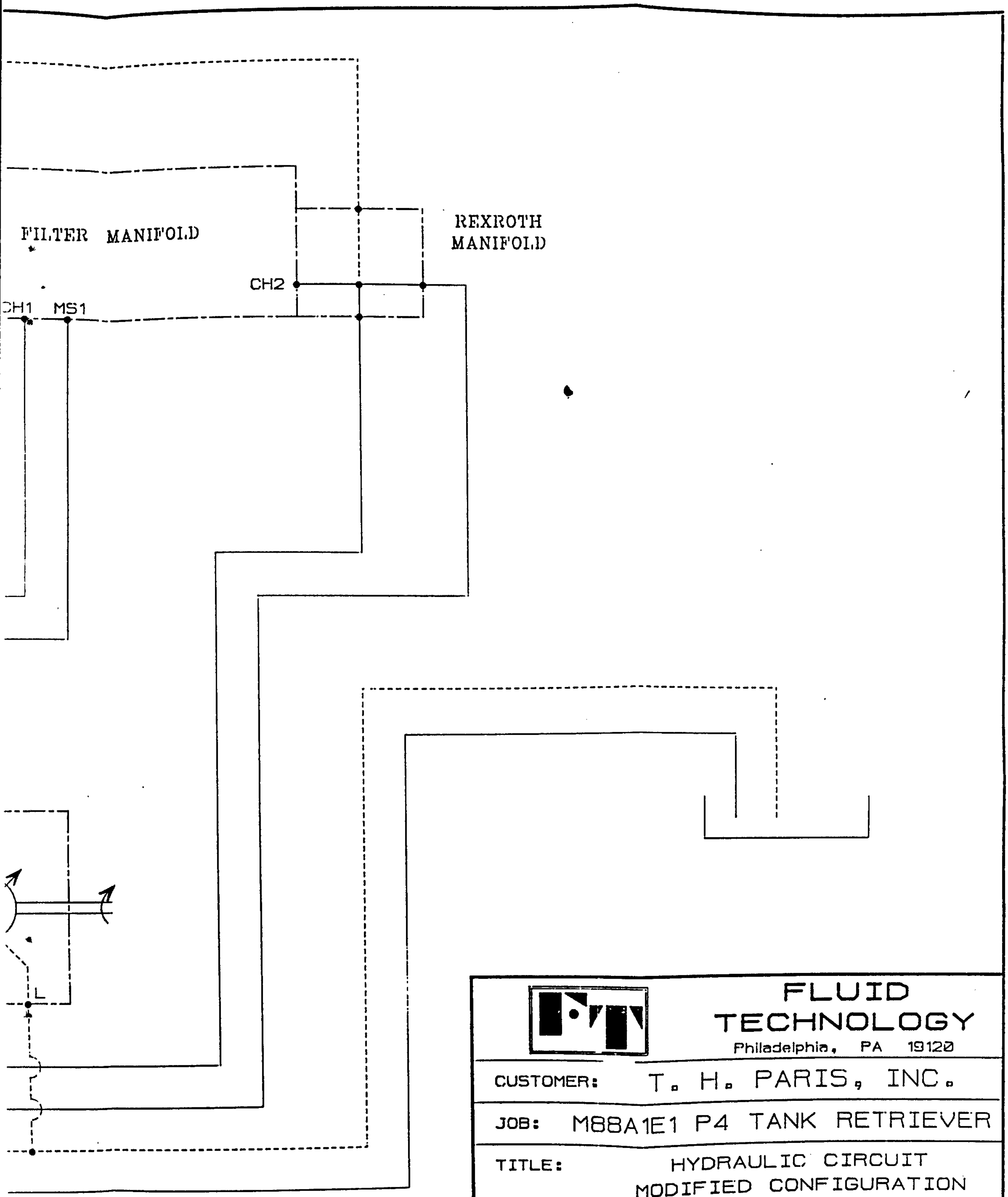
LOCATION
'OR HOSE ASSEMBLY INFORMATION
?EXROTH

FLUID TECHNOLOGY
Philadelphia, PA 19120

CUSTOMER:	T. H. PARIS, INC.
JOB:	M88A1E1 P4 TANK RETRIEVER
TITLE:	HYDRAULIC CIRCUIT MODIFICATION FITTINGS SPECIFICATIONS
DWG: B 960	DATE: 04/08/91
	BY: DP

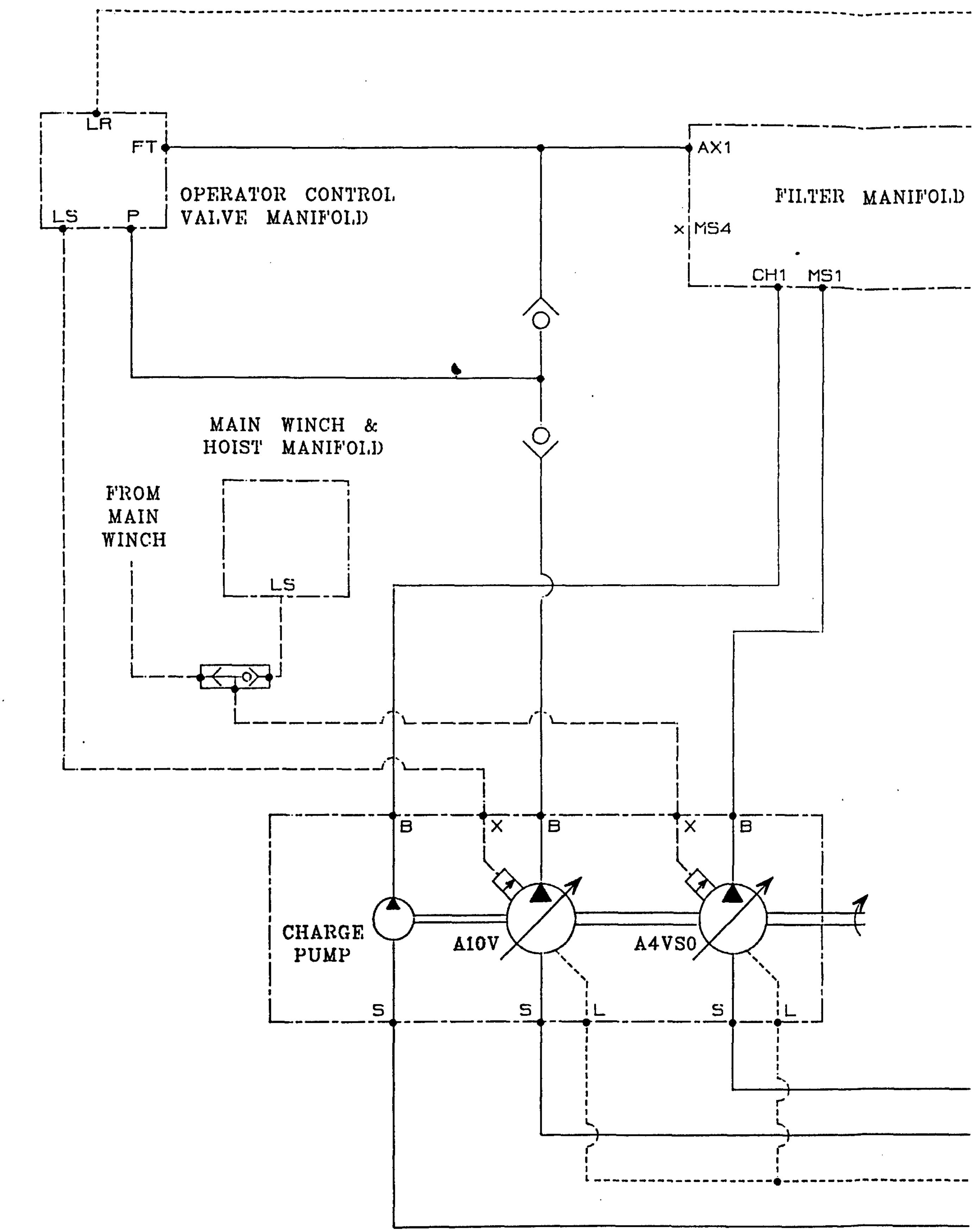
FITTING	DESCRIPTION	
F1	BRANCH TEE	SAE "O" RING
F2	ADAPTER - STR.	SAE "O" RING
F3	HEX UNION	SAE 37 DEG.
F4	ADAPTER - STR.	SAE "O" RING
F5	SWIVEL NUT RUN TEE	SAE 37 DEG.
F6	SWIVEL NUT BRANCH TEE	SAE 37 DEG.
F7	ADAPTER - STR.	SAE "O" RING
F8	HEX HEAD PLUG	SAE "O" RING
F9	ELBOW - 45 DEG.	SAE "O" RING
F10	ADAPTER - STR.	SAE "O" RING
F11	ADAPTER - STR.	SAE "O" RING
F12	ADAPTER - STR.	SAE "O" RING
F13	ADAPTER - STR.	SAE "O" RING
F14	ADAPTER - STR.	SAE "O" RING

NOTES: REFER TO DWG. B 959 FOR FITTING LOCATION
REFER TO DWGS. B 957 & B 958 FOR HOSE A
M1, M2 & M3 ARE MANIFOLDS BY REXROTH

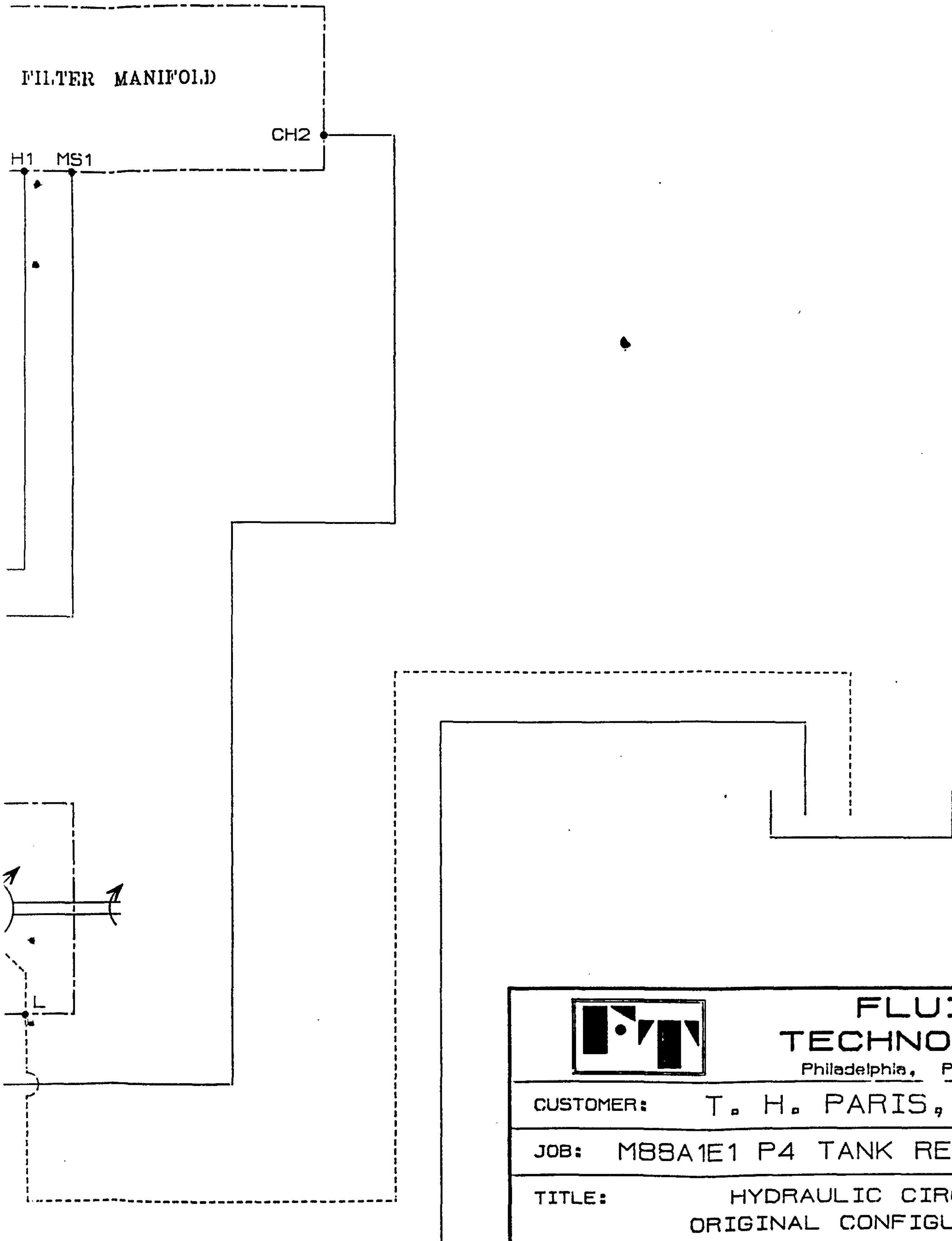


FLUID TECHNOLOGY
Philadelphia, PA 19120

CUSTOMER:	T. H. PARIS, INC.
JOB:	M88A1E1 P4 TANK RETRIEVER
TITLE:	HYDRAULIC CIRCUIT MODIFIED CONFIGURATION
DWG:	B 961
DATE:	04/11/91
BY:	DP

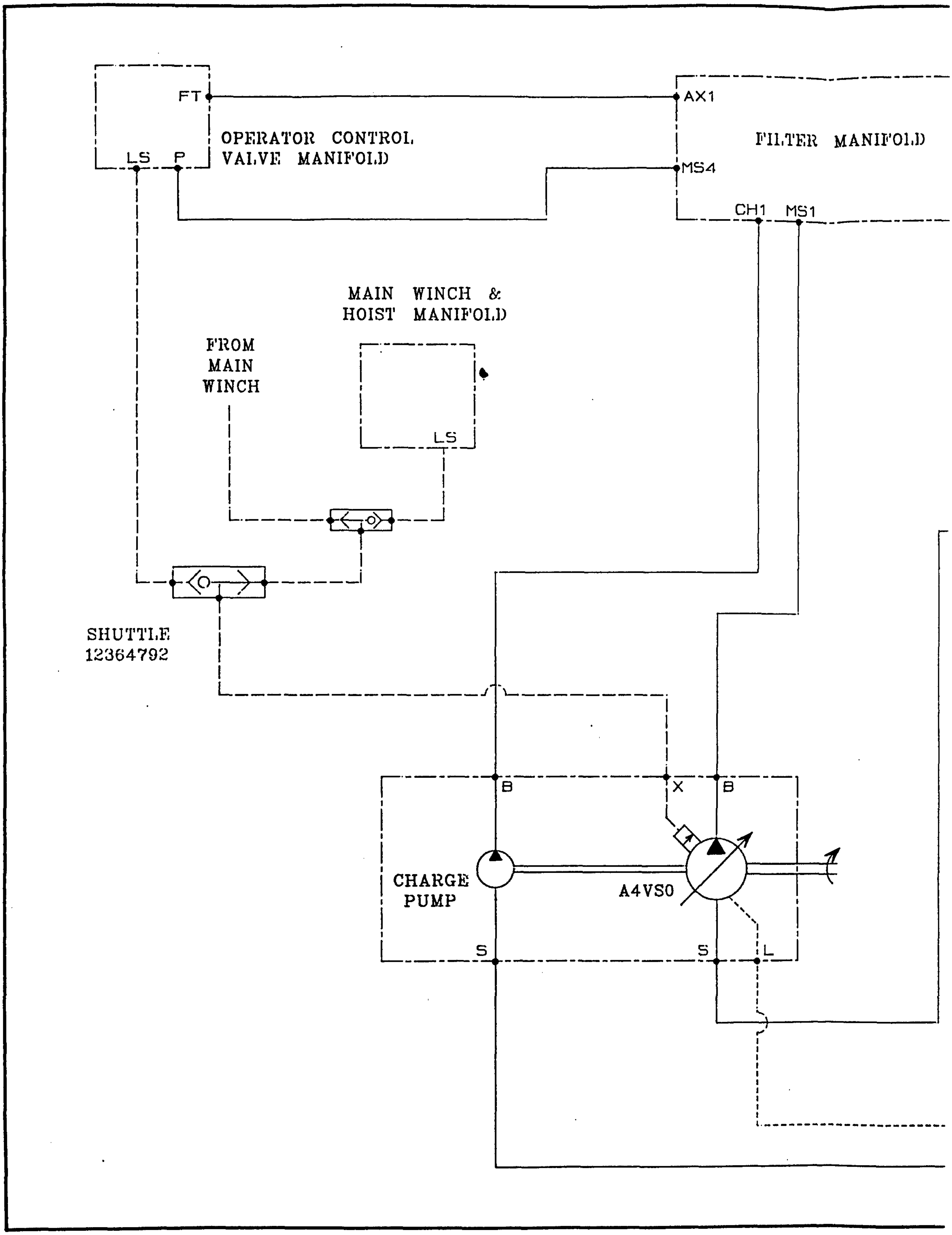


NOTE: LINES NOT ILLUSTRATED REMAIN UNCHANGED



FLUID TECHNOLOGY
Philadelphia, PA 19120

CUSTOMER:	T. H. PARIS, INC.
JOB:	M88A1E1 P4 TANK RETRIEVER
TITLE:	HYDRAULIC CIRCUIT ORIGINAL CONFIGURATION
DWG: B 962	DATE: 04/11/91
	BY: DP


FT
LS
P
OPERATOR CONTROL VALVE MANIFOLD
AX1
MS4
FILTER MANIFOLD
CH1
MS1
MAIN WINCH & HOIST MANIFOLD
FROM MAIN WINCH
LS
SHUTTLE 12364792
B
X
B
CHARGE PUMP
A4VSO
S
S
L

13-Apr-91

TION

	MFG.	PART #	REF. DWGS.
	REXROTH	AA10VSO45DFR/30L-PKC62K02	B 957, B 959
	REXROTH	SK43-A53-0409-C	*
Y PUMP	REXROTH	SK43-A53-0407-B	B 957, B 959, *
PUMP	REXROTH	SK43-A53-0406-B	B 957, B 959, *
	REXROTH	SK43-A53-0408-C	B 957, B 959, *
	REXROTH	S15A/12	B 957, B 959
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 957, B 958
			B 959, B 960
			B 959, B 960
			B 959, B 960
			B 959, B 960
			B 959, B 960
			B 959, B 960
			B 959, B 960
			B 959, B 960
			B 959, B 960
- 24			B 957, B 958
- 20			B 957, B 958
- 12			B 957, B 958

FLUID TECHNOLOGY, INC. PHILADELPHIA, PA

FTI JOB #: 106

TACOM - M88A1E1 P4 TANK RETRIEVER MODIFICATION

BILL OF MATERIALS

ITEM	QTY.	REF.	DESCRIPTION	
1	1	A10V	AUXILIARY PUMP	REXR
2	1		PUMP FOOT BRACKET	REXR
3	1	M2	PRESSURE PORT MANIFOLD FOR AUXILIARY PUMP	REXR
4	1	M1	PRESSURE PORT MANIFOLD FOR CHARGE PUMP	REXR
5	1	M3	ADAPTER MANIFOLD ON FILTER MANIFOLD	REXR
6	2		CHECK VALVE	REXR
7	1	H1	HOSE ASSEMBLY	
8	1	H3	HOSE ASSEMBLY	
9	1	H5A	HOSE ASSEMBLY	
10	1	H4	HOSE ASSEMBLY	
11	1	H4A	HOSE ASSEMBLY	
12	1	H32A	HOSE ASSEMBLY	
13	1	H35A	HOSE ASSEMBLY	
14	1	H10A	HOSE ASSEMBLY	
15	1	H10B	HOSE ASSEMBLY	
16	1	HLR	HOSE ASSEMBLY	
17	1	F1	BRANCH TEE	
18	1	F2	ADAPTER - STR	
19	1	F3	HEX UNION	
20	1	F4	ADAPTER - STR	
21	1	F5	SWIVEL NUT RUN TEE	
22	1	F6	SWIVEL NUT BRANCH TEE	
23	6	F7, F10-F14	ADAPTER - STR	
24	1	F8	HEX HEAD PLUG	
25	1	F9	ELBOW 45 DEG.	
26	1	H4A-b	SAE SPLIT FLANGE KIT - CODE 61 SAE - 24	
27	1	H4A-a	SAE SPLIT FLANGE KIT - CODE 61 SAE - 20	
28	1	H5A-b	SAE SPLIT FLANGE KIT - CODE 61 SAE - 12	

NOTES:

SAE FLANGES REMOVED FOR SYSTEM MODIFICATION WHICH MAY BE USED IN NEW CONFIGURATION ARE NOT LISTED

* REFER TO REXROTH DRAWINGS - DWG # IDENTICAL TO P/N

Designations: AA10VS045DRG/30L-PKC62K02 (Center Pump)

= Pressure Port 3/4" SAE 3000 PSI Flange (Code 61)
= Suction Port 1-1/2" SAE 3000 PSI Flange (Code 61)
= Leakage Port (7/8-14UNF-2B) SAE-10 Straight Thread Port
= Pilot Pressure Port (7/16-20UNF-2B) SAE-4 Straight Thread Port

cal Data: AA10VS045DRG/30L-PKC62K02 (Center Pump)

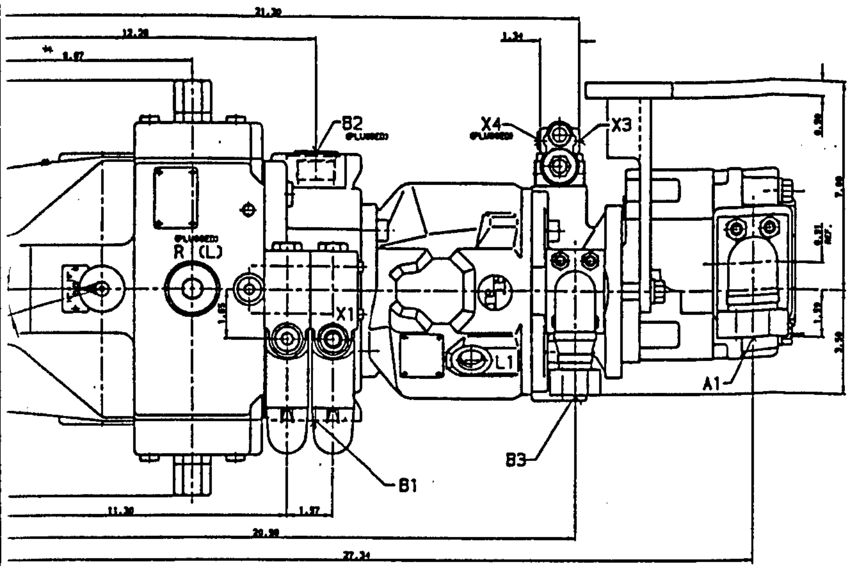
acement Vg = 2.75in³/rev. (Max.)
ng Pressure P = 3500 psi Compensator Setting
Drive Speed nmax = 2800 rpm (Flooded suction)
y Pressure = 480 psi

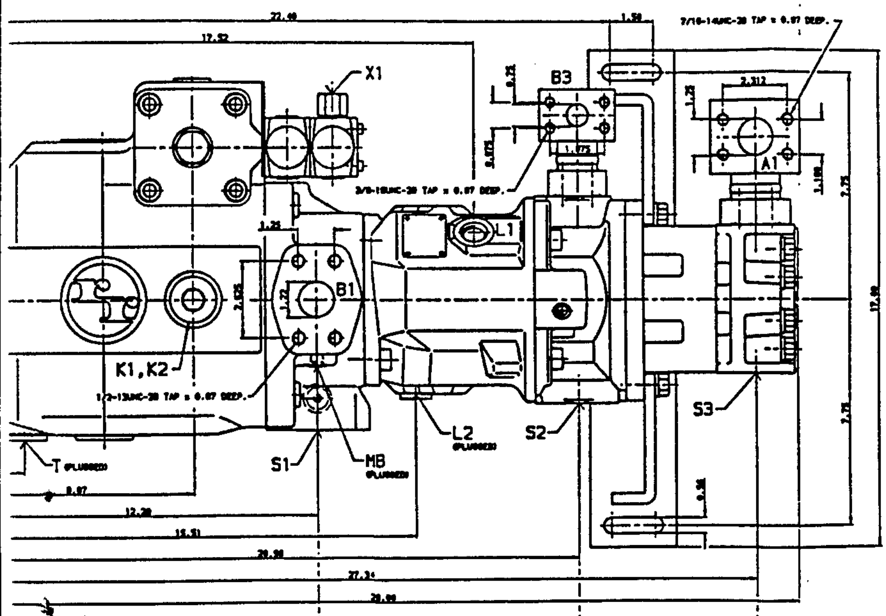
Port Designations: S30S-24HJ1-5-L (Rear Pump)

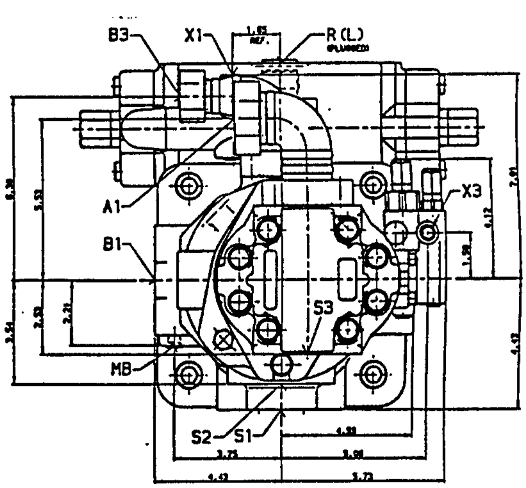
A1 = Pressure Port 1-1/4" SAE 3000 PSI Flange (Code 61)
S2 = Suction Port 1-1/2" SAE 3000 PSI Flange (Code 61)

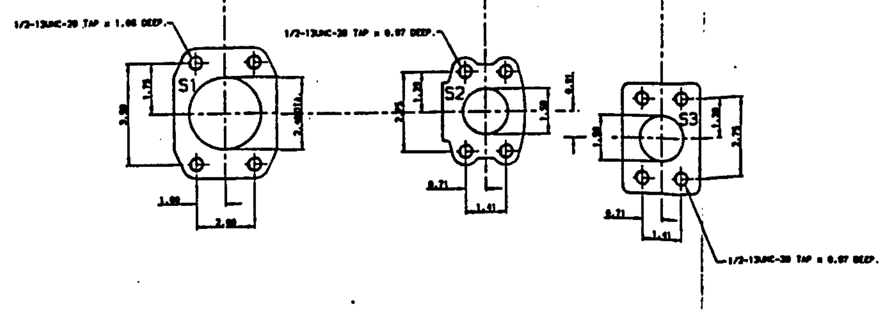
Technical Data: S30S-24HJ1-5-L (Rear Pump)

Displacement Vg = 2.74in³/rev.
Working Pressure P = 3000 psi (Continuous Pressure)
Max. Drive Speed nmax = 3000 rpm









APR 15 1991

REXROTH

OUTLINE DRAWING
TRIPPLE PUMP

SK43-A53-0402-E-2

Port Designations: AA4VSO125DFR/10L-PSD63K05 (Front Pump)

B1	= Pressure Port	1-1/4" SAE 6000 PSI Flange (Code 62)
B2	= Additional Press.	(1-5/8-12UN-2B) SAE-20 Straight Thread Port
S1	= Suction Port	2-1/2" SAE 3000 PSI Flange (Code 61)
K1,K2	= Flushing Port	(1-5/16-12UN-2B) SAE-16 Straight Thread Port
T	= Oil Discharge	(1-5/16-12UN-2B) SAE-16 Straight Thread Port
R(L)	= Return Aeration	(1-5/16-12UN-2B) SAE-16 Straight Thread Port
MB	= Outlet Pressure	(7/16-20UNF-2B) SAE-4 Straight Thread Port
MS	= Inlet Pressure	(9/16-18UNF-2B) SAE-6 Straight Thread Port
X1	= Control Pressure	(7/16-20UNF-2B) SAE-4 Straight Thread Port

Technical Data: AA4VSO125DFR/10L-PSD63K05 (Front Pump)

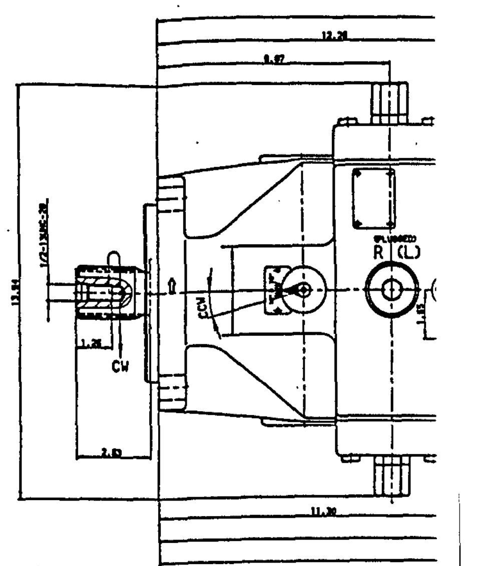
Displacement	Vg	=	7.60in³/rev. (Max.)
Compensator Press	P	=	4500 psi (Nominal Pressure)
Max. Drive Speed	nmax	=	2600 rpm (With Boosted Suction)
Standby Pressure		=	380 psi

Port Designations: AA10VSO4

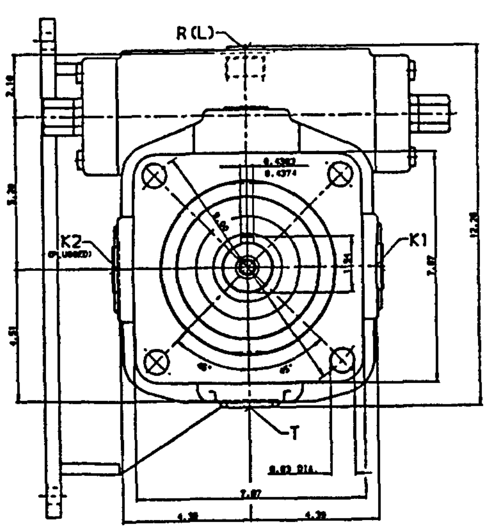
B3	= Pressure Port
S2	= Suction Port
L1,L2	= Leakage Port
X3,X4	= Pilot Pressure Port

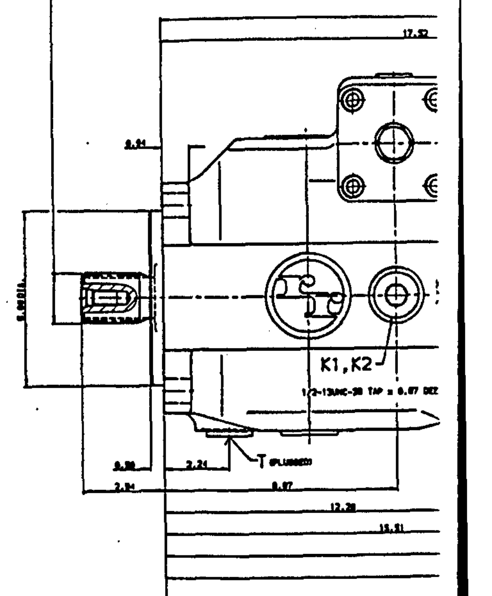
Technical Data: AA10VSO4

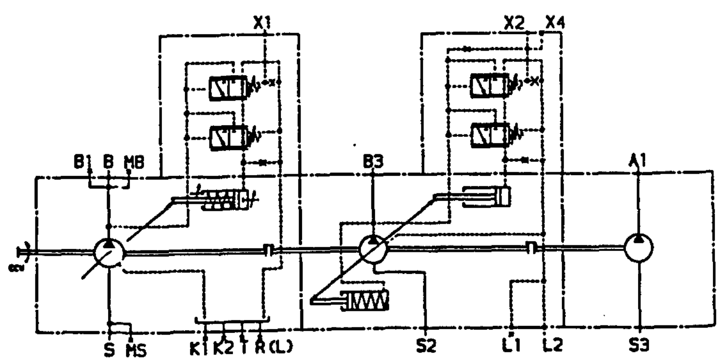
Displacement	Vg	=	2
Working Pressure	P	=	3
Max. Drive Speed	nmax	=	2
Standby Pressure		=	4



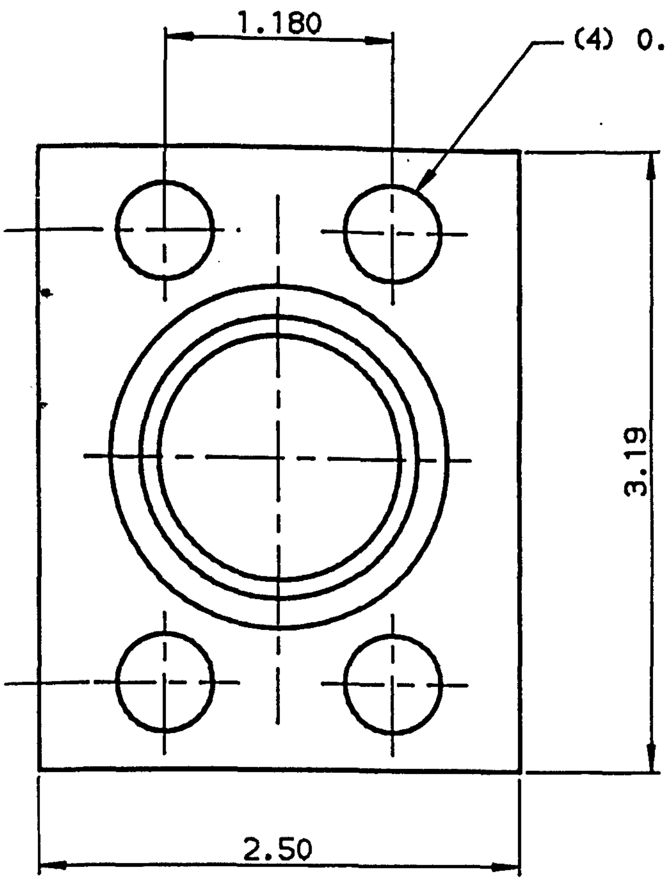
1 3/4 SPLINE SIZE, 30° PRESSURE ANGLE, 13 TEETH, 8/16 PITCH FLAT ROOT, -SIDE FIT TOL.- CLASS 5; ANSI B.92.1a - 1976







Hydraulic Circuit Diagram



QTY	DESCRIPTION	MODEL / SIZE	POS	PART NUMBER	MATERIAL	REMARKS
1	1-1/4" SCH.80 x .50 LG.		4	804237	A53	CUT TO SUIT
1	1-1/4" SCH.80 BUTTWELD 90°		3	804983	A120	SHORT RAD.
1	1-1/4" SAE BUTTWELD FLANGE		2	811479	1018	CODE 61
1	1-1/4" SAE BUTTWELD FLANGE		1	811480	1018	CODE 61

Rev	Date	Name	Remarks
6			
5			
4			
3			
2			
1			

UNLESS OTHERWISE SPECIFIED TOLERANCES ARE IN INCHES
2 PLACE DECIMALS ±.06
3 PLACE DECIMALS ±.020
ANGULAR DIMENSIONS ±1°

REXROTH
WORLDWIDE HYDRAULICS
BETHLEHEM, PA 18017

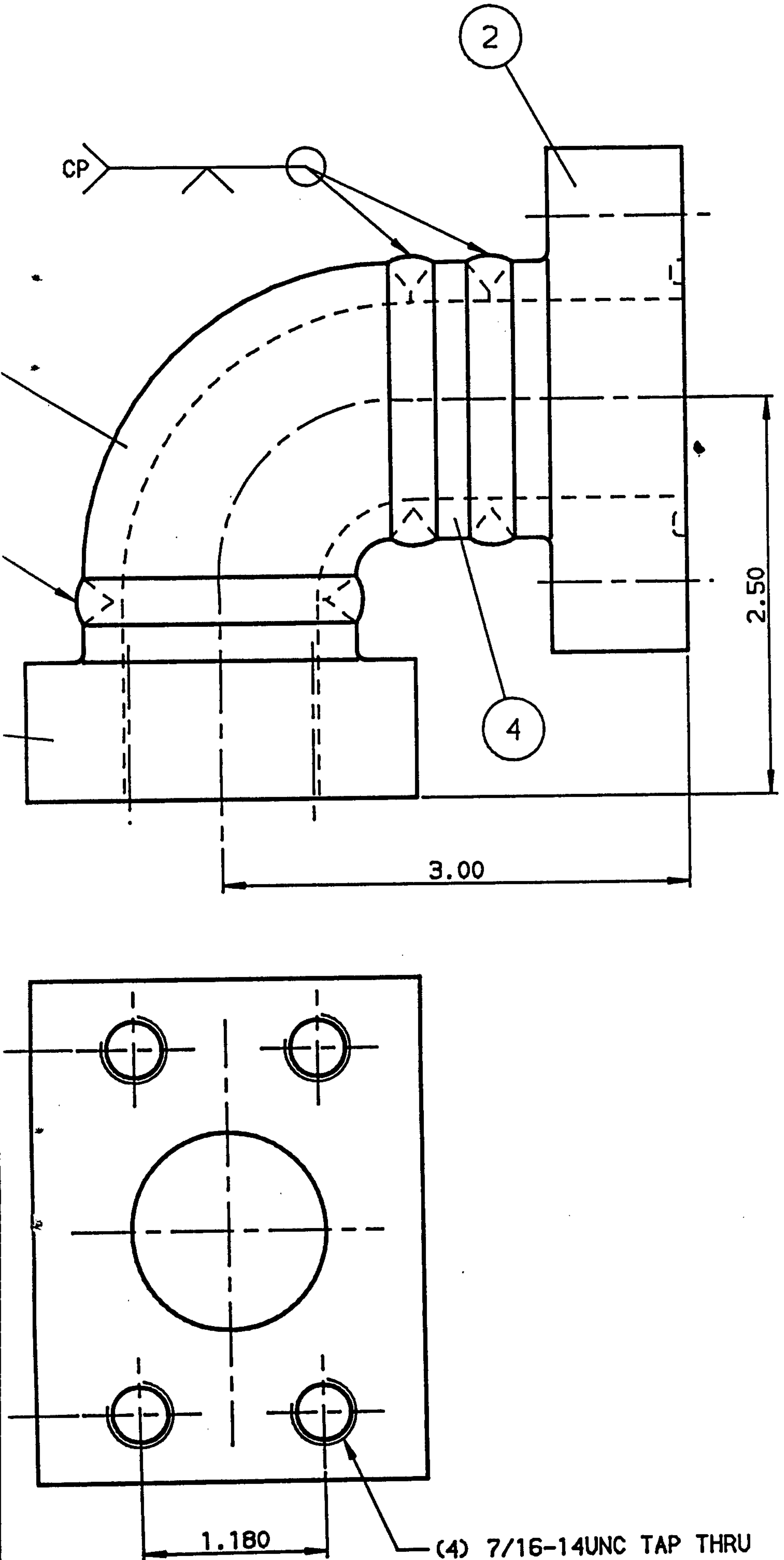
DWN BY:	Date:	Material:	Scale:
FERENCZY	03-20-91	SEE SCHEDULE	1:1

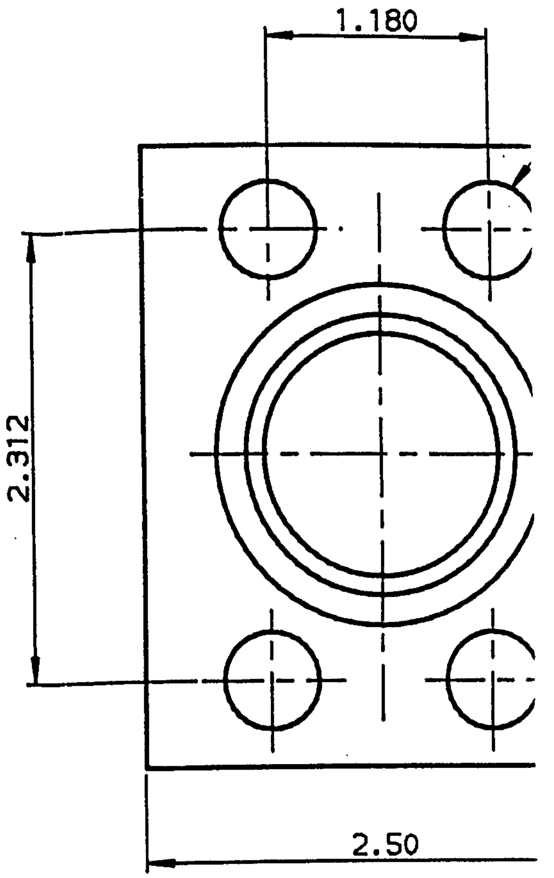
Supersedes: *

Sheet 1 of 1

Description:
WELDED PIPE ASSEMBLY
FOR M88A1E1 GEAR PUMP

Drawing No. / Part No.
SK43-A53-0406-B-0



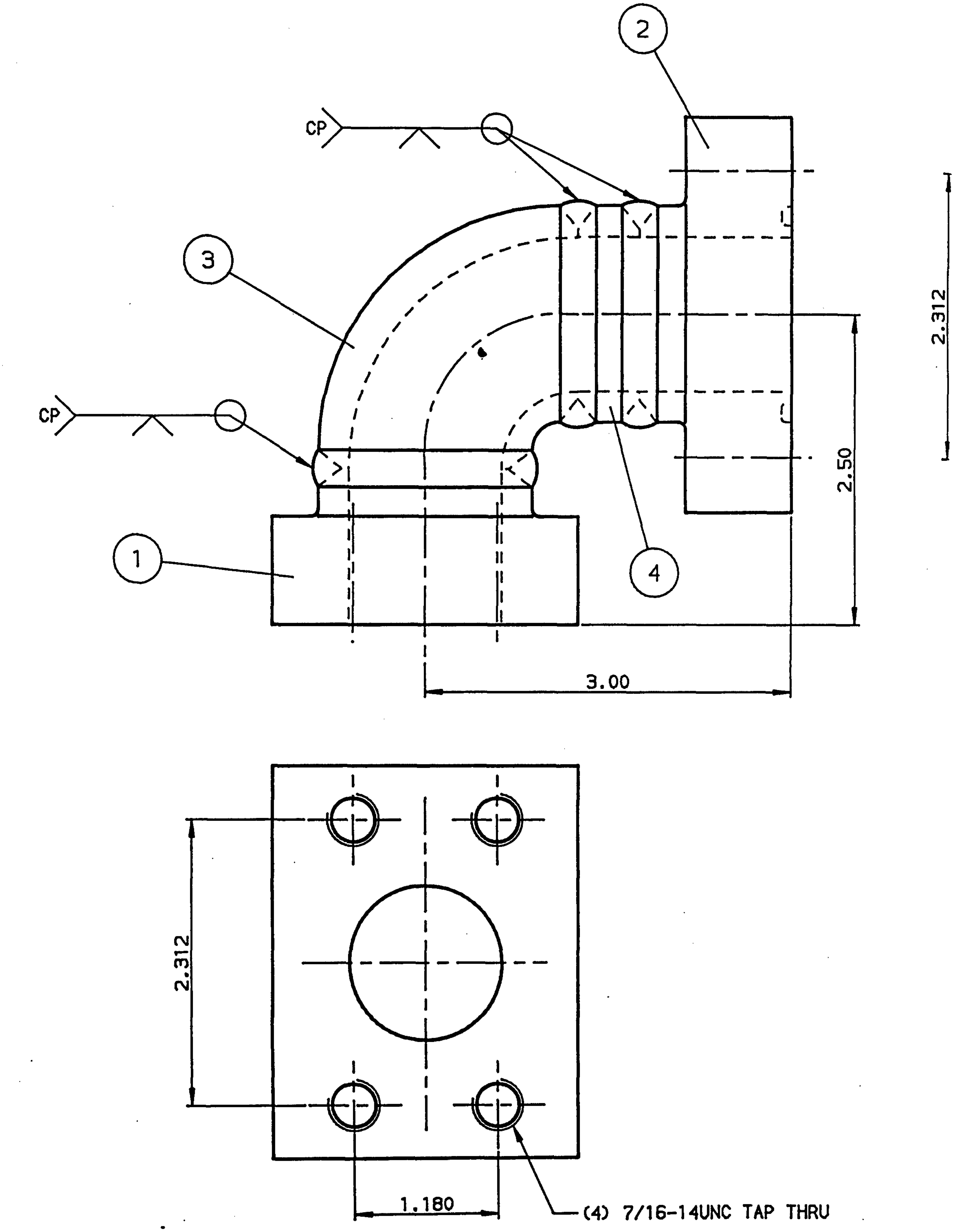


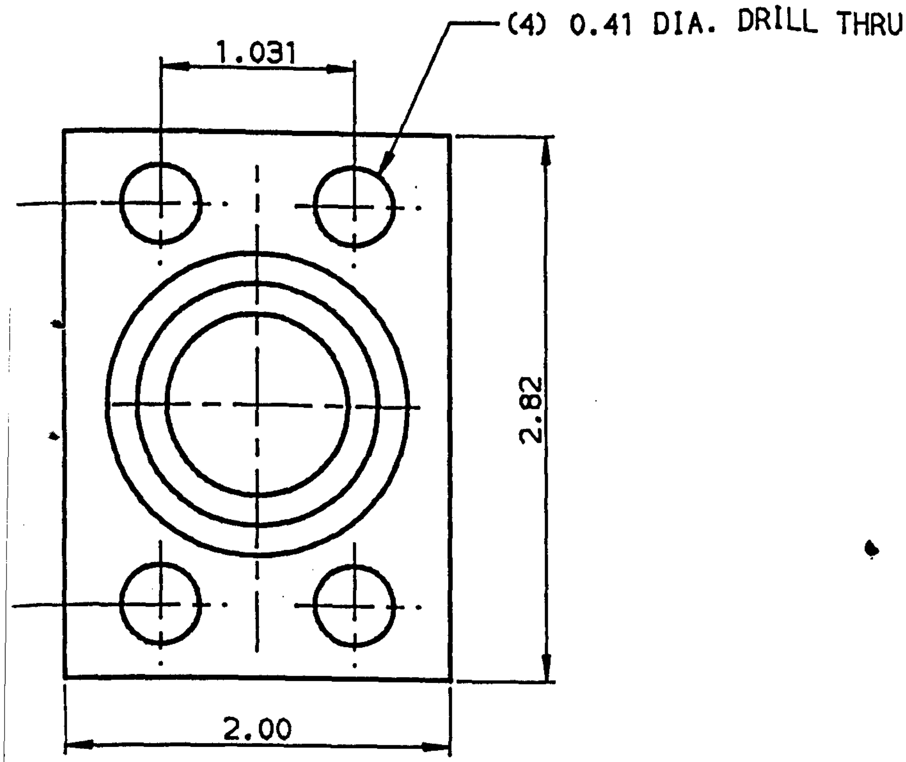
1	1-1/4" SCH.
1	1-1/4" SCH.
1	1-1/4" SAE
1	1-1/4" SAE
QTY	DESCRIPTION

6		
5		
4		
3		
2		
1		
Rev	Date	Name

DWN BY: FERENCZY	Date: 03-20-91
Supersedes: *	

CAD-ORIGINAL - REV


2
CP
3
CP
2.312
2.50
1
4
3.00
2.312
1.180
(4) 7/16-14UNC TAP THRU



1	REDUCER SK43-A53-0410-B-0	4	*	1018	1" x 3/4"
1	1" SCH.80 BUTTWELD 90°	3	804982	A120	LONG RAD.
1	1" SAE BUTTWELD FLANGE	2	811477	1018	CODE 61
1	3/4" SAE BUTTWELD FLANGE	1	811476	1018	CODE 61
QTY	DESCRIPTION / MODEL / SIZE	POS	PART NUMBER	MATERIAL	REMARKS

Rev	Date	Name	Remarks
6			
5			
4			
3			
2			
1	03-28-91	FERENCZY	REVISED AS BUILT

UNLESS OTHERWISE SPECIFIED
TOLERANCES ARE IN INCHES
2 PLACE DECIMALS ±.06
3 PLACE DECIMALS ±.020
ANGULAR DIMENSIONS ±1°

REXROTH
WORLDWIDE HYDRAULICS
BETHLEHEM, PA 18017

DWN BY:	Date:	Material:	Scale:
FERENCZY	03-20-91	SEE SCHEDULE	1:1

Supersedes: *

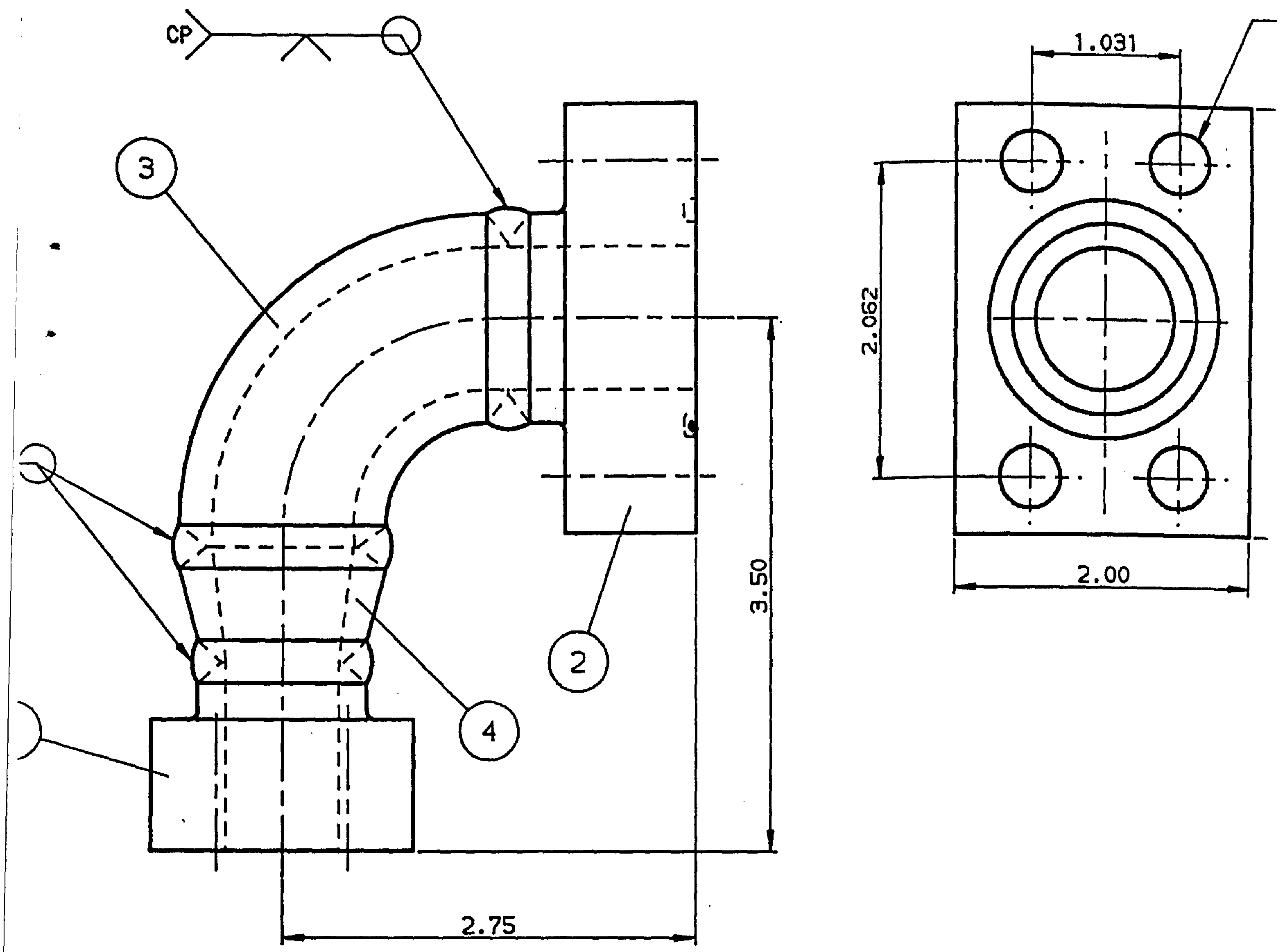
Sheet 1 of 1

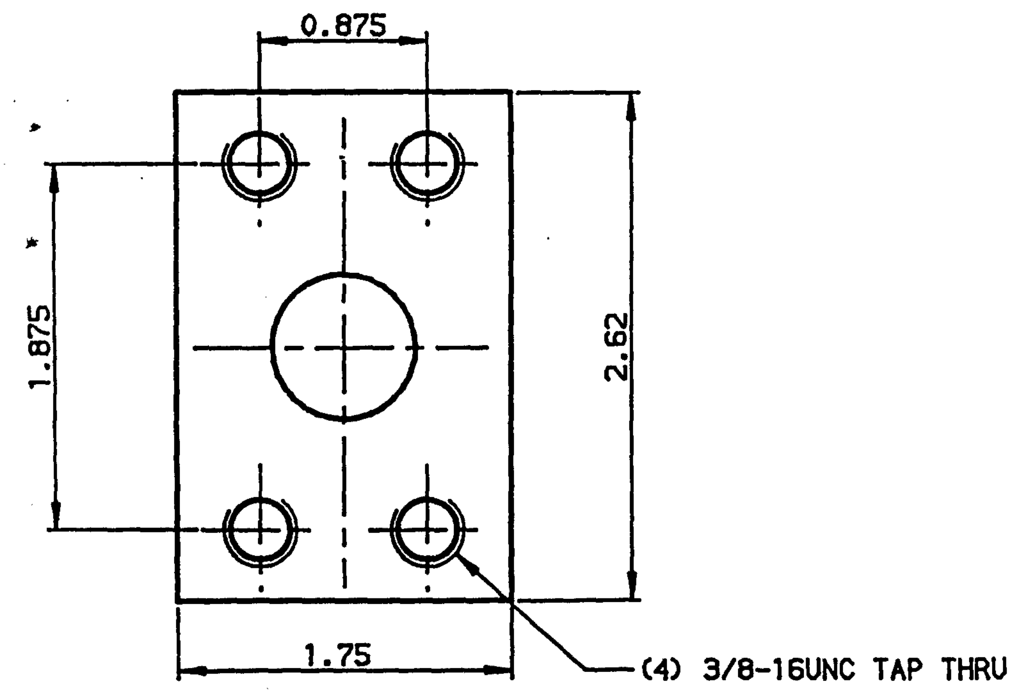
Description:
WELDED PIPE ASSEMBLY
M88A1E1 A10V PUMP

Drawing No. / Part No.
SK43-A53-0407-B-1

0407-B-1

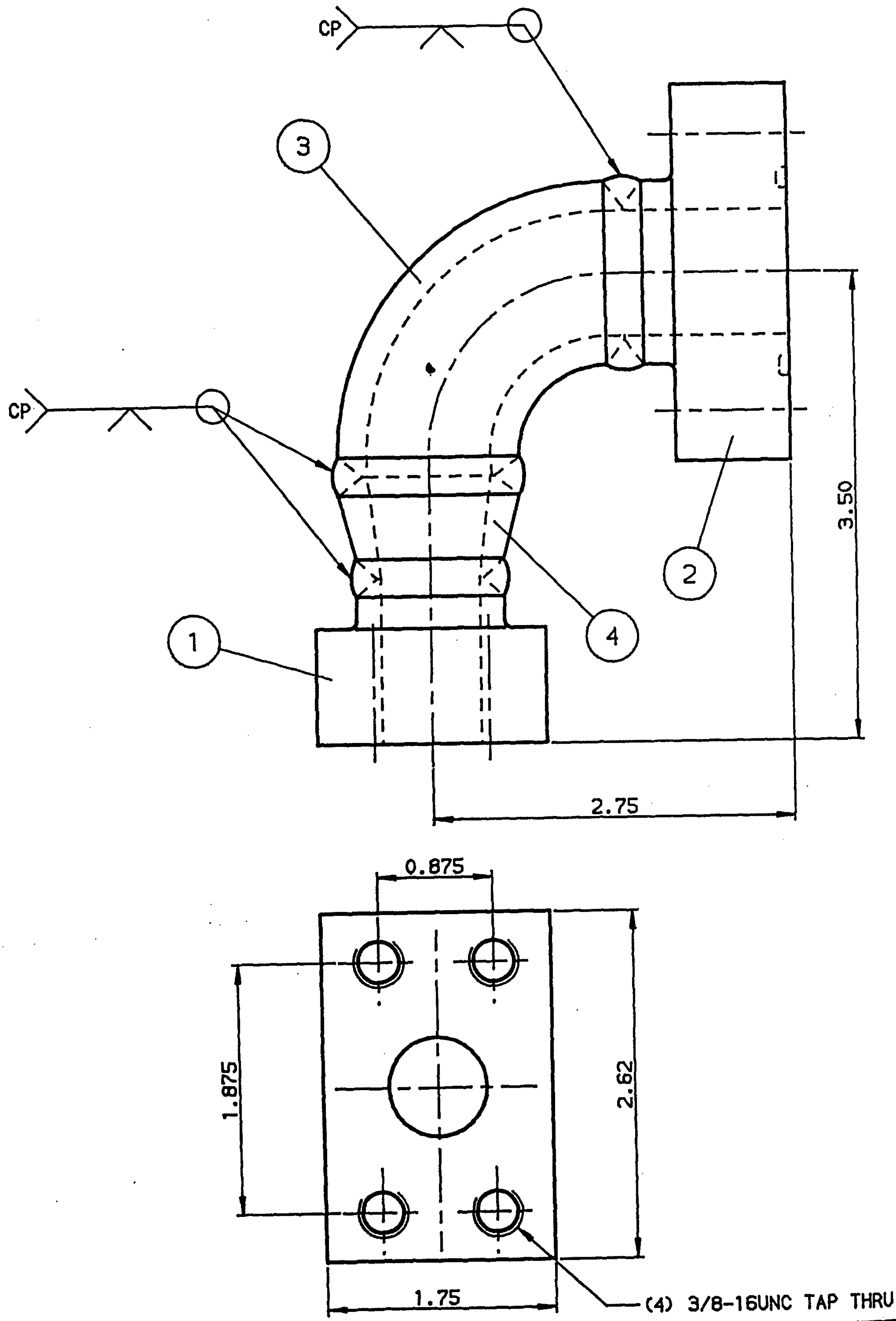
CAD-ORIGINAL: REVISIONS ONLY POSSIBLE THRU CAD





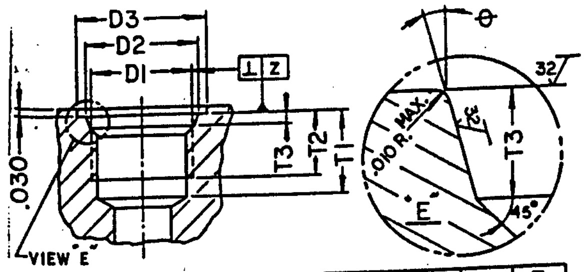
1	REDUCER SK43	
1	1" SCH.80 BU	
1	1" SAE BUTTW	
1	3/4" SAE BUT	
QTY	DESCRIPTION	
6		
5		
4		
3		
2		
1	03-28-91	FERENCZY
Rev	Date	Name
DWN BY: FERENCZY	Date: 03-20-91	Mat
Supersedes: *		

CAD-ORIGINAL, REVIS



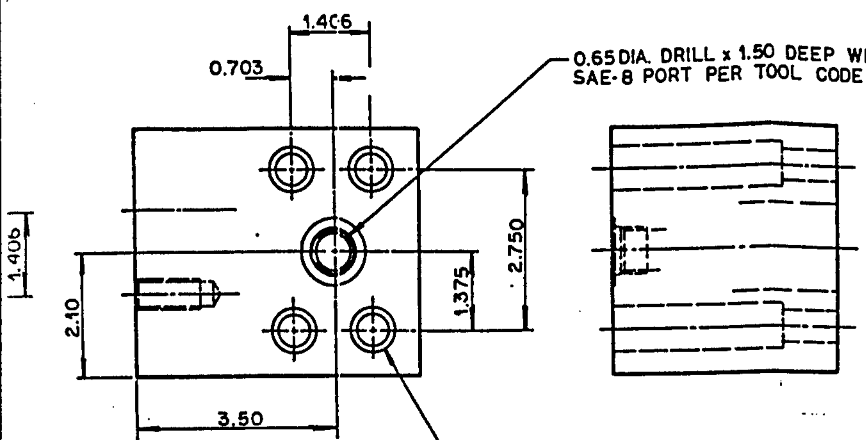
2 062

0.75 DIA. DRILL x 1.75 DEEP WITH SAE-12 PORT PER TOOL CODE "a".



TOOL CODE	SAE SIZE	D1	D2	D3 MIN.	T1	T2 MIN.	T3	θ	Z
	4	7/16-20	.487	.828	.547	.454	.093	12°	
	5	1/2-20	.550	.906					
	6	9/16-18	.616	.969	.609	.500	.097		
b	8	3/4-16	.811	1.188	.688	.562	.100		.002
	10	7/8-14	.942	1.344	.781	.656			
a	12	1-1/16-12	1.148	1.625				15°	
	14	1-3/16-12	1.273	1.765			.130		
	16	1-5/16-12	1.398	1.910	.906	.750			
	20	1-5/8-12	1.713	2.270					.004
	24	1-7/8-12	1.962	2.560			.132		
	32	2-1/2-12	2.587	3.480					

(4) 0.42 DIA. DRILL x 1.37 DEEP WITH 1/2-13 NC TAP x 1.09 DEEP.

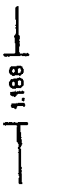


0.65 DIA. DRILL x 1.50 DEEP WITH SAE-8 PORT PER TOOL CODE "b".

DIA. DRILL x 3.75 DEEP.

(4) 0.53 DIA. DRILL THRU WITH 0.78 DIA. C. BORE x 3.00 DEEP.

IA. DRILL x 2.00 DEEP.



NOTES:

- MANIFOLD IS TO BE CLEANED AND DEBURRED.
- FINISHES OTHER THEN SPECIFIED ARE TO BE 250 OR BETTER.
- PORT IDENTIFICATIONS AND PART NUMBER, INDICATED BY ⊂⊃, ARE TO BE STAMPED AS MARKED. LETTERS AND NUMBERS ARE TO BE A MINIMUM OF 1/4 INCH HIGH.
- TOLERANCES: 2 PLACES .01 3 PLACES .005

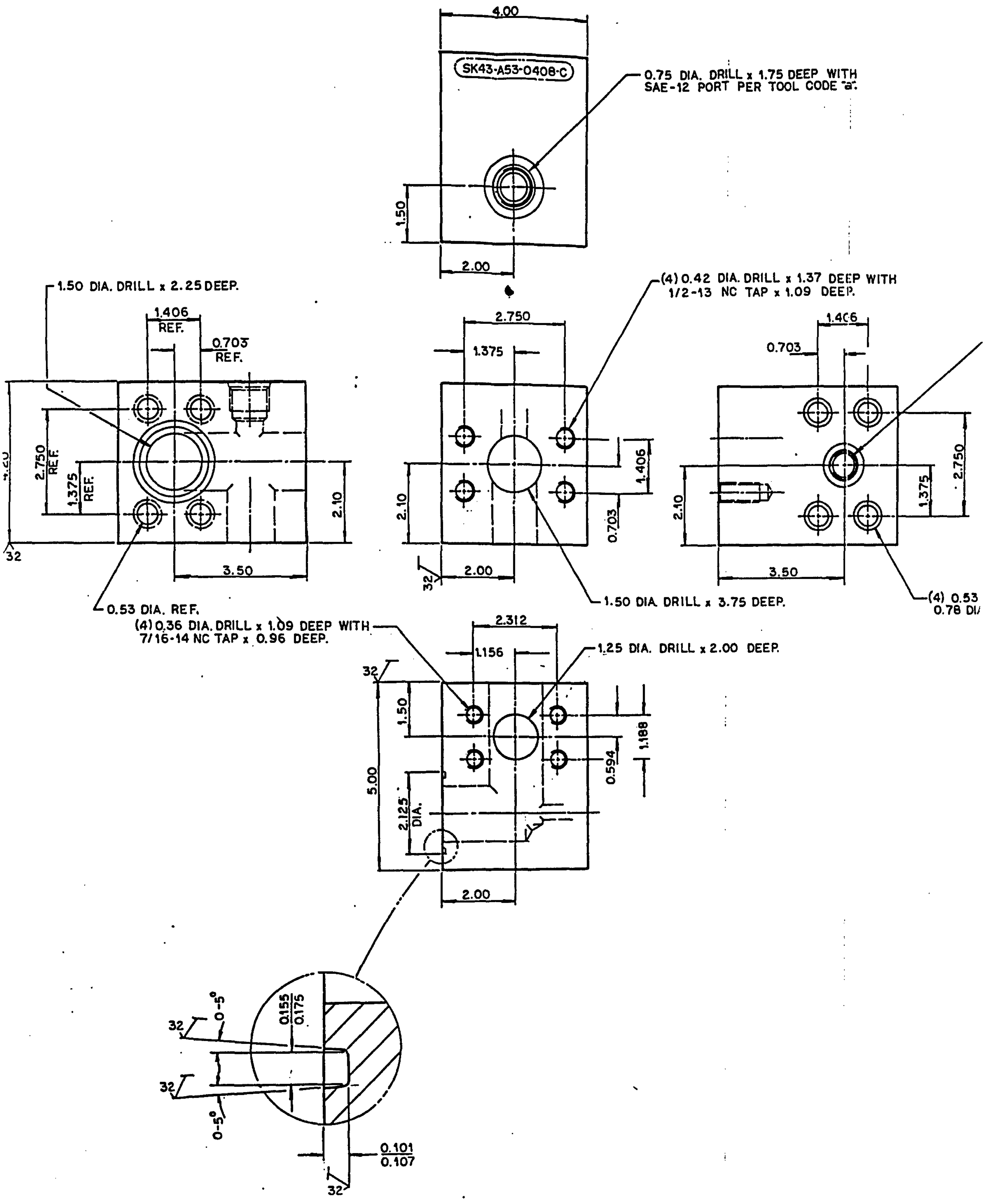
QTY	DESCRIPTION	MODEL / SIZE	POS	PART NUMBER	MATERIAL	REMARKS
g						
f						
e						
d						
c						
b						
a	4-4-91	MF	1	SAE-8 WAS SAE-4		
Index	Date	Name	Change No.	Remarks		

TOLERANCE IN INCHES

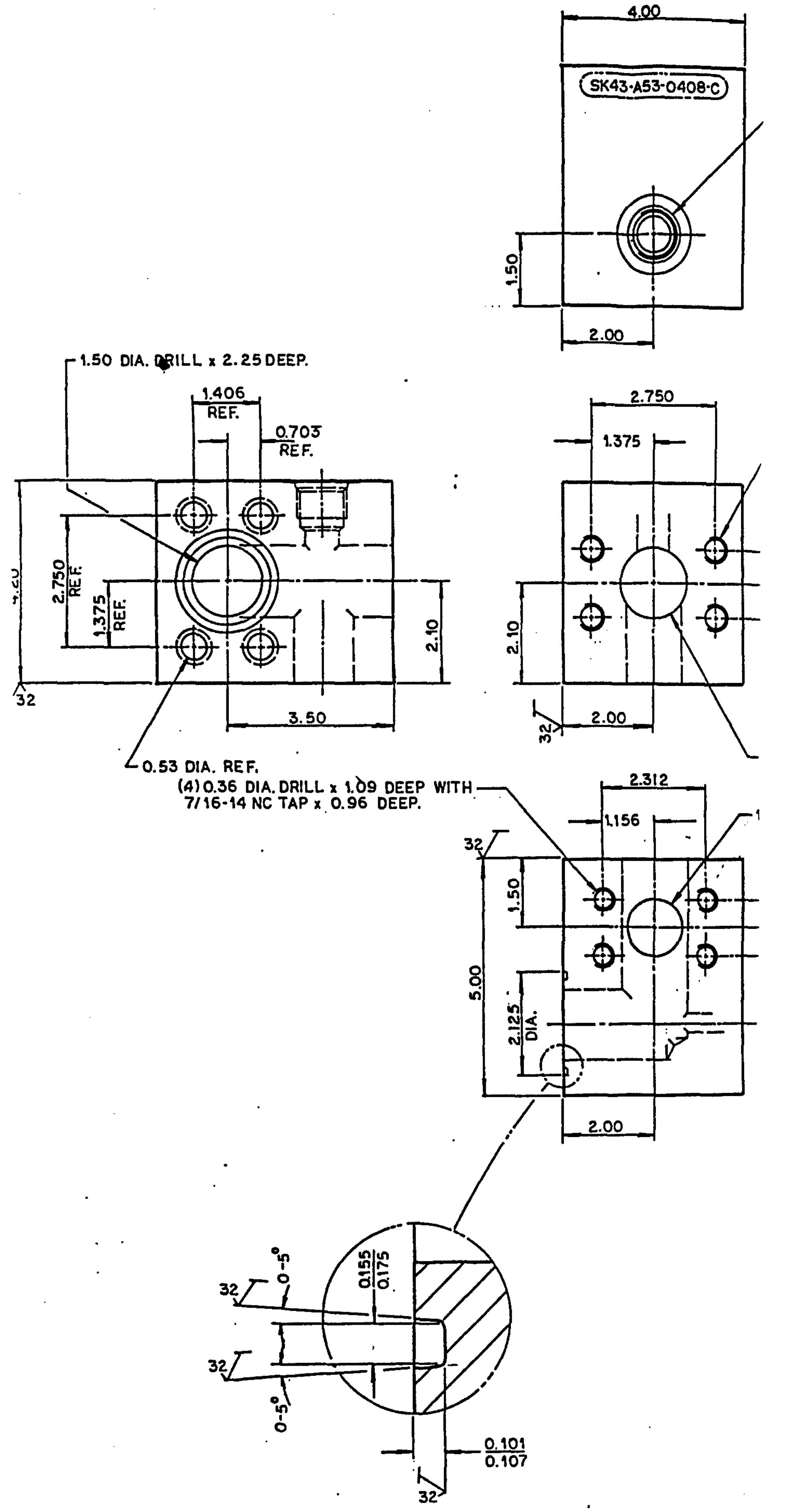
21.000	±.025		
13.800	±.020	432	±.250
7.800	±.015	288	±.188
4.500	±.012	144	±.156
2.800	±.010	72	±.125
1.500	±.008	48	±.063
1.000	±.006	24	±.031
0.600	±.005		
Size To:	Finish Machine	Size To:	Steel Fab

1991	DATE	NAME	MODEL No.			Tolerance	Dimension
Drawn	3-21	MF	Material: DUCTILE IRON	Sheet No. 1	Scale: 1:2	DESCRIPTION M88A1E1 SUCTION HEADER BLOCK	
Check			Finish Size: 5.0 x 4.2 x 4.0	01 1 Sheets		Part Code SK43-A53-0408-C-1	Part No.
App'd							

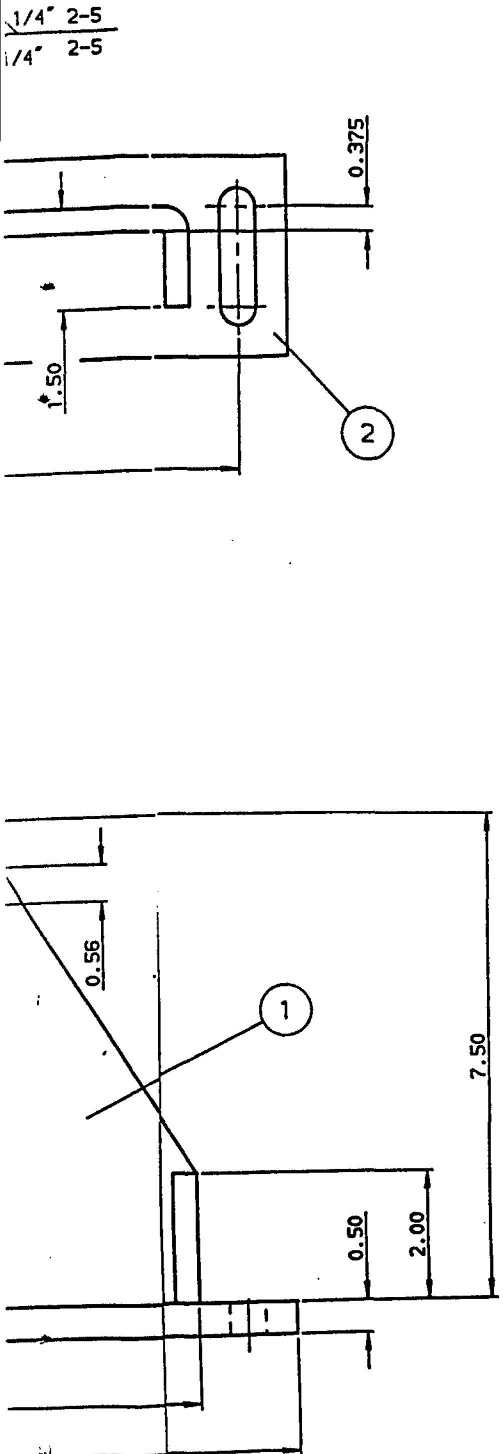
REXROTH Corporation



4.00
SK43-A53-0408-C
0.75 DIA. DRILL x 1.75 DEEP WITH SAE-12 PORT PER TOOL CODE "a".
1.50
2.00
1.50 DIA. DRILL x 2.25 DEEP.
1.406 REF.
0.703 REF.
2.750 REF.
1.375 REF.
2.10
3.50
32
0.53 DIA. REF.
(4) 0.42 DIA. DRILL x 1.37 DEEP WITH 1/2-13 NC TAP x 1.09 DEEP.
2.750
1.375
1.406
0.703
2.10
2.00
1.50 DIA. DRILL x 3.75 DEEP.
1.406
0.703
2.750
1.375
2.10
3.50
(4) 0.53
0.78 DI
(4) 0.36 DIA. DRILL x 1.09 DEEP WITH 7/16-14 NC TAP x 0.96 DEEP.
2.312
1.156
1.25 DIA. DRILL x 2.00 DEEP.
1.50
5.00
2.125 DIA.
1.188
0.594
2.00
0-5°
0.155
0.175
0.101
0.107



4.00
SK43-A53-0408-C
1.50
2.00
1.50 DIA. DRILL x 2.25 DEEP.
1.406
REF.
0.703
REF.
2.750
REF.
1.375
REF.
2.10
3.50
32
0.53 DIA. REF.
2.750
1.375
2.10
2.00
32
(4) 0.36 DIA. DRILL x 1.09 DEEP WITH
7/16-14 NC TAP x 0.96 DEEP.
2.312
1.156
32
1.50
5.00
2.125
DIA.
2.00
0.155
0.175
0-5°
32
32
0-5°
0.101
0.107
32



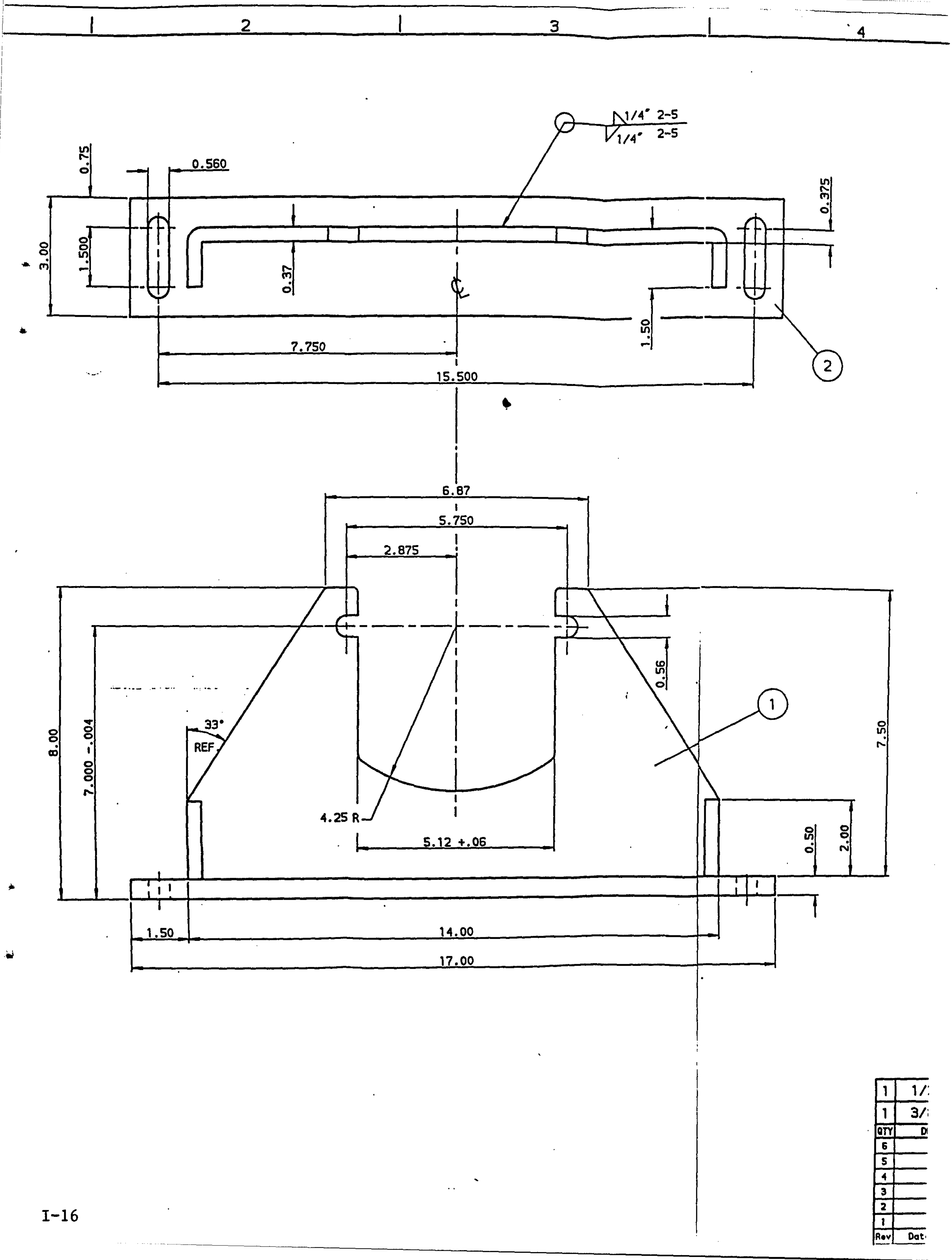
1	1/2" PLATE	17.00 X 3.00	2	*	1020HRS	*
1	3/8" PLATE	17.00 x 7.50	1	*	1020HRS	BEND
QTY	DESCRIPTION	MODEL / SIZE	POS	PART NUMBER	MATERIAL	REMARKS

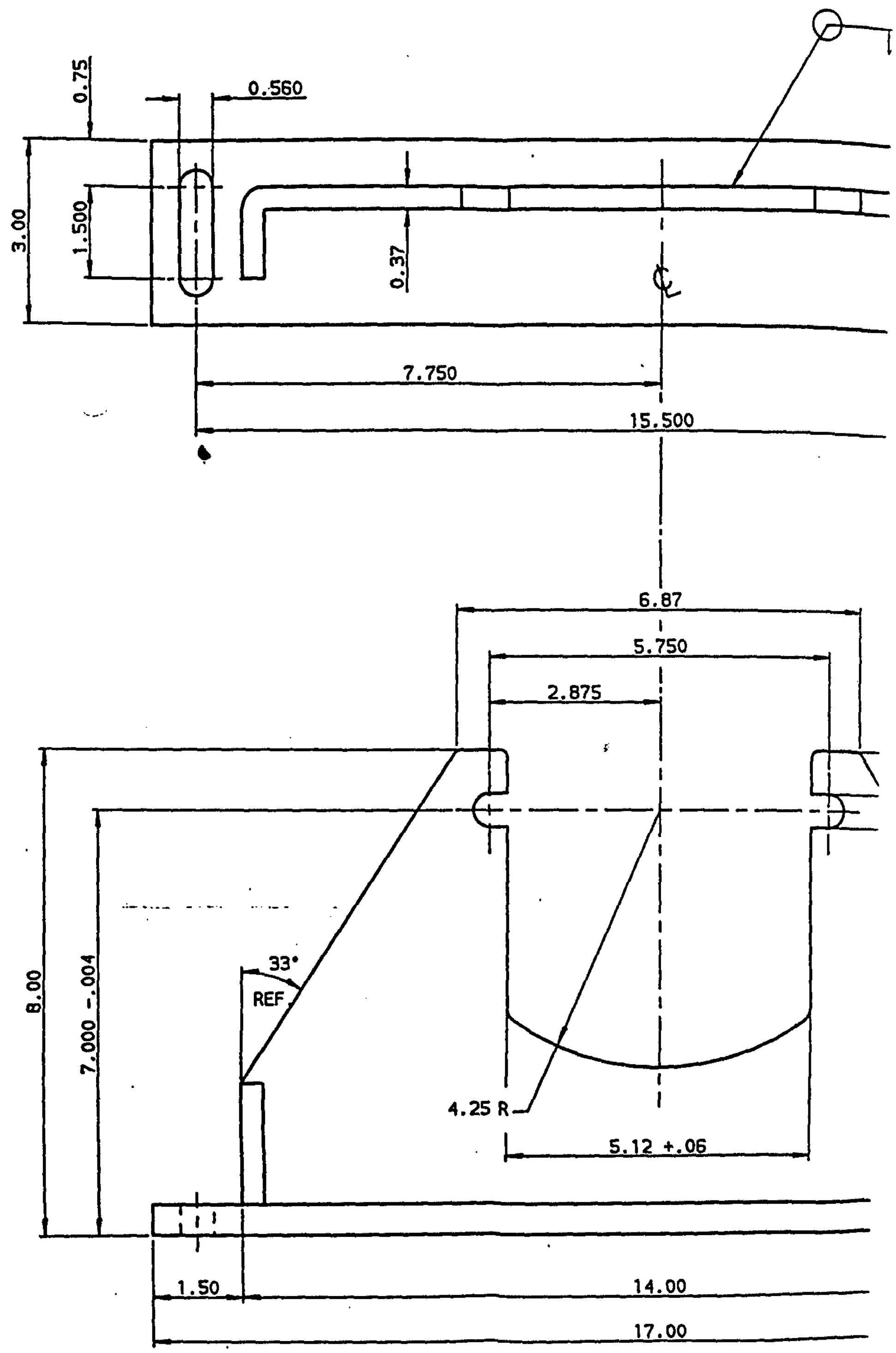
Rev	Date	Name	Remarks
6			
5			
4			
3			
2			
1			

UNLESS OTHERWISE SPECIFIED
TOLERANCES ARE IN INCHES
2 PLACE DECIMALS ±.06
3 PLACE DECIMALS ±.020
ANGULAR DIMENSIONS ±1°

REXROTH
WORLDWIDE HYDRAULICS
BETHLEHEM, PA 18017

Description:
TRIPPLE PUMP SUPPORT BRACKET
M88A1E1 TANK RECOVERY VEHICLE


2
3
4
1/4" 2-5
1/4" 2-5
0.75
0.560
0.375
3.00
1.500
0.37
1.50
7.750
15.500
2
6.87
5.750
2.875
0.56
1
33°
REF
8.00
7.000 -.004
7.50
4.25 R
5.12 +.06
0.50
2.00
1.50
14.00
17.00
QTY
Rev


0.75
0.560
3.00
1.500
0.37
7.750
15.500
6.87
5.750
2.875
33°
REF
8.00
7.000 -.004
4.25 R
5.12 +.06
1.50
14.00
17.00

APPENDIX J

Parts List

M88A1E1 PROTOTYPE #4

List of parts added resulting from Summer 90 - Winter 91 Hydraulic System Test at Aberdeen Proving Grounds (APG)

SYSTEM	PART	Qty	MAKE/MODEL
1. Main winch	Double-Pilot-Operated Check Valve (DPOCV) (also installed in vehicle #P5)	2	Snap Tite Inc. # CAD 10-N6S-25A
2. Main winch	Fittings for DPOCV	4 4	SAE-6 to 37 degree Flair (male)
3. Main winch	Hose assemblies for DPOCV	2	1/4" dia - 18" long
4. Level wind	Level wind cam bracket (thru bore core to .628" for snug fit)	1	APG fabricated 1/4" 1040 mild steel (1 welded assembly)
5. Level wind	Retainer brackets for center shaft	2	

6. Directional Control Valve (DCV) — (The following changes were made by Mr. George See of BMY)

   a. Removed 3/8" check ball, replaced with 7/16" ball

   b. Shuttle valve added to DCV "G" port

   c. Load sense hose assembly 1/4" line - to shuttle valve on DCV

   d. A "tee" connection was made at the winch motor; included is a 37 degree female on run, a 37 degree male flair on run and 37 degree male on branch.

7. Main winch — Reservoir drain was tied to the main winch motor drain; it was tee'd in but is now capped. The drain presently goes directly to reservoir.

SYSTEM	PART	Qty	MAKE/MODEL
8. 3-Pump System	Pump # SK43-A53-0402-E-O	1	Rexroth 4310-NSN

SYSTEM	PART	Qty	MAKE/MODEL
9. 3-Pump System	Manifold block	1	Rexroth FBM
10. 3-Pump System	Manifold block	1	Rexroth A10V-MB
11. 3-Pump System	Manifold block	1	Rexroth S30S-MB
12. 3-Pump System	Manifold bracket	1	Rexroth S30S-FB
13. 3-Pump System	Check valve 3/4" male fitting	2	Rexroth S15A1.0/12-SO
14. 3-Pump System	3/8" needle valve assembly	2	Rexroth NVA38
15. 3-Pump System	1/4" needle valve assembly	2	Rexroth NVA14
16. 3-Pump System	Code 61 Split Flanges		Aeroquip
	a. # 107-74446-24	5 pr	1 1/2"
	b. # 107-74446-20	2 pr	1 1/4"
	c. # 107-74446-16	2 pr	1"
	d. # 107-74446-12	1 pr	3/4"
17. 3-Pump System	Adapters		Aeroquip
	a. # 203003-16-16S	1	1" tee, 37 degree flare w/SAE "O" ring 1" boss on branch
	b. # 202702-10-8S	1	5/8" O-ring boss to 1/2" 37 degree flare male
	c. # 2027-4-4S	1	Straight 37 degree male flare 1/4"-1/4" union

SYSTEM	PART	Qty	MAKE/MODEL
	d. # 202702-4-4S	1	Straight 1/4" SAE O-ring boss to 37 degree male flare 1/4"
	e. # 203102-12-12S	1	3/4" tee 37 degree flare male on run, male on branch, female swivel on run
	f. # 203101-12-12S	1	3/4" tee 37 degree male on run, 37 degree female swivel branch
	g. # 202702-12-12S	1	Straight 3/4" O-ring boss 3/4" 37 degree male
	h. # 700598-12S	1	3/4" SAE O-ring boss plug
	i. # 2061-12-12S	1	45 degree SAE O-ring boss to 37 degree flare male 3/4" to 3/4"
	j. # 202702-12-12S	1	3/4" SAE O-ring to 3/4" 37 degree male flare (straight)
	k. # 221501-16-8S	1	37 degree -16 female 37 degree -8 male flair
	l. # 107-74446-20	1	Code 61 1 1/4" split flange hardware
18. 3-Pump System	Hoses and fittings		Aeroquip

Dimensions	Type	Fittings on Ends
a. 25" CUT -24	FC318	45 degree -24 Code 61 split flange, straight -24 code 61 split flange
b. 29" CUT -16	2781	Straight -16 code 61 split flange 90 degree -16 code 61 split flange

Dimensions	Type	Fittings on Ends
c. 40" CUT -12	2781	Straight 37 degree female flair -12 90 degree -12 code 61 split flange
d. 18" CUT -24	FC318	Straight -24 code 61 split flange 45 degree -24 code 61 split flange
e. 27" CUT -20	FC318	90 degree -20 code 61 split flange straight -24 code 61 split flange
f. 29" CUT -8	2781	Straight 37 degree female flair -16 90 degree 37 degree female flair -8
g. 40" CUT -4	2781	Straight 37 degree female flair -4 90 degree 37 degree female flair -4
h. 19" CUT -12	2781	Straight 37 degree female flair -12 45 degree 37 degree female -12
i. 24" CUT -12	2781	Straight 37 degree female flair -12 45 degree 37 degree female flair -12
j. 170" CUT -12	2781	Straight 37 degree female flair -12, 90 degree 37 degree female flair -1
k. 14" CUT -8	2781	Straight -8 1/2" 37 deg. female flair Straight -8 1/2" 37 deg. female flair
l. 32" CUT -16	2781	Straight -16 1" code 61 90 degree 1" straight code 61 -20

19. Operators manifold modified - no part number available; port "LR" added.

20. A bracket is available to support the new three-pump arrangement. It is not installed in vehicle #4; however, it will be shipped with the vehicle to BMY.

21. The BMY hook block for the hoist has been replaced with:

Johnson 30 Ton block
3/4" wire rope specification
Design factor 4
Model 30D14RTAB
Serial# S9-331
ASS.# 470234024

APPENDIX K

Oil Cooler

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M88A1E1 HYDRAULIC OIL TEMPERATURE ANALYSIS

Present and past testing of the M88A1E1 hydraulic systems has shown that typical work cycles cannot be completed without the oil temperature exceeding maximum allowable levels for safe operation.

To eliminate the problem, an oil cooler is required. A system temperature analysis was made to understand both how and where the temperatures were being affected. Upon analyzing the results, it was possible to arrive at a preliminary sizing for a cooler. Maximum temperatures were used to arrive at the cooler sizing.

The temperature and oil cooler analysis contain the following information:

I. System temperature analysis - a theoretical calculation of final system temperatures.

II. Determination of maximum hydraulic oil heat gain.

III. Preliminary cooler selection.

IV. Theoretical calculation of oil cooler sizing.

I. Theoretical Calculation of Final System Temperature for Test Runs 79-81

SUMMARY
(+ = Gains, - = Losses)

a. Starting Temperature	+ 162.0°F
b. Main Pump/Winch Motor Inefficiency	+ 118.08°F
c. Third Pump/Aux Motor Inefficiency	+ 8.42°F
d. Main Pump (during Aux Pump Operation)	+ 3.34°F
e. Charge Pump Inefficiency	+ 1.17°F
f. Loss Through Insulation	- 5.53°F
g. Piping Convection and Radiation	- 1.14°F
h. Reservoir Conduction	- 23.15°F
Calculated Theoretical Temperature	+ 263.91 °F
Actual Measured Temperature	+ 236.0 °F

Percent Difference From Theoretical = 11.8%

The following pages contain a detailed breakdown of each temperature gain and loss for the one-hour cycle based on data for runs 79-81, for aux/main winching.

a. Starting Temperature <u>+162°F</u>

b. The Main Pump and Main Winch motor both operate with identical flow rates and pressures. Since they both run for the same length of time and have the same estimated efficiency rating of 85%, the heat generated by both is theoretically assumed to be equal.

Main pump theoretical flow rate = 55 gpm
Main pump is a piston pump = 85% efficient (15% loss)

During a winch cycle the system pressure varies per wrap of the cable. Pressure variation affects horsepower (hp), therefore:

Bare Drum	Flow = 55 gpm; Pressure = 4280 psi
2nd Layer	Flow = 56 gpm; Pressure = 3330 psi
3rd Layer	Flow = 56 gpm; Pressure = 3030 psi

<u>Theoretical hp</u> (gpm x psi x constant)

$$hp_{\text{bare drum}} = 55 \times 4280 \times .000583 = 137 \text{ hp}$$
$$hp_{\text{2nd layer}} = 56 \times 3330 \times .000583 = 108 \text{ hp}$$
$$hp_{\text{3rd layer}} = 56 \times 3030 \times .000583 = 98.9 \text{ hp}$$

<u>hp Loss</u> (efficiency loss x hp)

$$Loss_{\text{bare drum}} = .15 \times 137 = 20.6 \text{ hp}$$
$$Loss_{\text{2nd layer}} = .15 \times 108 = 16.2 \text{ hp}$$
$$Loss_{\text{3rd layer}} = .15 \times 99 = 14.8 \text{ hp}$$

<u>Mass of Oil</u>

Total volume of oil = 101 gal. (per BMY)
Reservoir = 75 gal.

$m = V$ $\qquad \rho$ = density for oil = 53.19 lb./ft^3
V = volume = 101 gal.

$$m = 101 \text{ gal.} \times 53.19 \text{ lb./ft}^3 \times 231 \text{ in}^3\text{/gal} \times 1 \text{ ft}^3\text{/1728 in}^3 = 718.2 \text{ lb.}$$

<u>Heat Gain</u>

$$Q_{\text{bare drum}} = 20.6 \text{ hp} \times \frac{2545 \text{ BTU/hr}}{1 \text{ hp}} = 52{,}527 \text{ BTU/hr}$$

$$Q_{\text{2nd layer}} = 16.2 \text{ hp} \times \frac{2545 \text{ BTU/hr}}{1 \text{ hp}} = 41{,}229 \text{ BTU/hr}$$

$$Q_{3rd\ layer} = 14.8\ hp \times \frac{2545\ BTU/hr}{1\ hp} = 37{,}666\ BTU/hr$$

Temperature Increase (T = Q/mc, specific heat of oil = .509 BTU/lb-°F)

$$T_{bare\ drum} = \frac{52{,}427\ BTU/hr.}{718.2\ lb. \times .509\ BTU/lb\text{-}°F} \times \frac{9}{60}\ hr = 21.51°F$$

$$T_{2nd\ layer} = \frac{41{,}229\ BTU/hr.}{718.2\ lb. \times .509\ BTU/lb\text{-}°F} \times \frac{9}{60}\ hr = 16.92°F$$

$$T_{3rd\ layer} = \frac{37{,}666\ BTU/hr.}{718.2\ lb. \times .509\ BTU/lb\text{-}°F} \times \frac{12}{60}\ hr = 20.61°F$$

Total = 59.04°F

Total heat gain for Main Pump and Winch Motor = **+118.08°F**

c. The Third Pump and Auxiliary Winch Motor both operate with identical flow rates and pressures. Since they both run for the same length of time and have the same estimated efficiency rating of 85%, the heat generated by both is theoretically assumed to be equal.

Third Pump/Aux Winch Motor theoretical flow rate = 21 gpm
Main pump is a piston pump = 85% efficient (15% loss)

During an aux winch cycle the system pressure varies per wrap of the cable. Pressure variation affects horsepower (hp), therefore:

Cable Out (co) Flow = 21 gpm; Pressure = 1140 psi
Cable Pull (cp) Flow = 21 gpm; Pressure = 3450 psi
Cable In (ci) Flow = 21 gpm; Pressure = 1290 psi

Theoretical hp (gpm x psi x constant)

$$hp_{co} = 21 \times 1140 \times .000583 = 14\ hp$$
$$hp_{cp} = 21 \times 3450 \times .000583 = 42\ hp$$
$$hp_{ci} = 21 \times 1290 \times .000583 = 16\ hp$$

hp Loss (efficiency loss x hp)

$$Loss_{co} = .15 \times 14 = 2.1\ hp$$
$$Loss_{cp} = .15 \times 42 = 6.3\ hp$$
$$Loss_{ci} = .15 \times 16 = 2.4\ hp$$

Heading: Heat Gain

$$Q_{co} = 2.1 \text{ hp} \times \frac{2545 \text{ BTU/hr}}{1 \text{ hp}} = 5{,}334 \text{ BTU/hr}$$

$$Q_{cp} = 6.3 \text{ hp} \times \frac{2545 \text{ BTU/hr}}{1 \text{ hp}} = 16{,}033 \text{ BTU/hr}$$

$$Q_{ci} = 2.4 \text{ hp} \times \frac{2545 \text{ BTU/hr}}{1 \text{ hp}} = 6{,}108 \text{ BTU/hr}$$

Temperature Increase (T = Q/mc)

$$T_{co} = \frac{5{,}334 \text{ BTU/hr.}}{718.2 \text{ lb.} \times .509 \text{ BTU/lb-°F}} \times \frac{3}{60} \text{ hr} = .73°\text{F}$$

$$T_{cp} = \frac{16{,}033 \text{ BTU/hr.}}{718.2 \text{ lb.} \times .509 \text{ BTU/lb-°F}} \times \frac{4}{60} \text{ hr} = 2.92°\text{F}$$

$$T_{ci} = \frac{6{,}108 \text{ BTU/hr.}}{718.2 \text{ lb.} \times .509 \text{ BTU/lb-°F}} \times \frac{2}{60} \text{ hr} = .56°\text{F}$$

Total = 4.21°F

Total heat gain for Main Pump and Winch Motor = **+8.42°F**

d. The Main Pump operates at 500 psi and is 85% efficient during Aux Pump operation.

Theoretical hp (gpm x psi x constant)

55 gpm x 500 psi x .000583 = 16 hp

hp Loss (efficiency loss x hp)

Loss = .15 x 16 = 2.4 hp

Heat Gain

$$Q = 2.4 \text{ hp} \times \frac{2545 \text{ BTU/hr}}{1 \text{ hp}} = 6{,}120 \text{ BTU/hr}$$

Temperature Increase (T = Q/mc)

$$T = \frac{6{,}120 \text{ BTU/hr.}}{718.2 \text{ lb.} \times .509 \text{ BTU/lb-°F}} \times \frac{12}{60} \text{ hr} = +3.34°\text{F}$$

e. The Charge Pump ran for the full one hour cycle The unit is a gear type pump and has an efficiency of 75%.

Theoretical hp (gpm x psi x constant)

21 gpm x 80 psi x .00583 = .98 hp

hp Loss (efficiency loss x hp)

Loss = .25 x .98 = .24 hp

Heat Gain

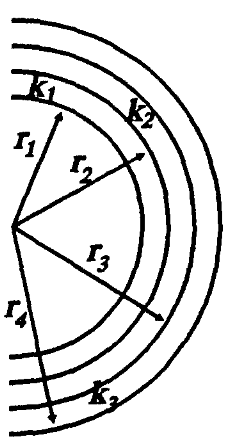
$$Q = .24 \text{ hp} \times \frac{2545 \text{ BTU/hr}}{1 \text{ hp}} = 624 \text{ BTU/hr}$$

Temperature Increase (T = Q/mc)

$$T = \frac{624 \text{ BTU/hr.}}{718.2 \text{ lb.} \times .509 \text{ BTU/lb-°F}} \times 1 \text{ hr} = \underline{+1.71°F}$$

f. Heat transmission through pipe insulation.

$$Q_{heat\ loss} = \frac{2\,L(T_2-T_1)}{\frac{1}{k_1}\ln(\frac{r_2}{r_1})+\frac{1}{k_2}\ln(\frac{r_3}{r_2})+\frac{1}{k_3}\ln(\frac{r_4}{r_3})}$$



Half Section of Hydraulic Piping

r_1 = 0.4275 in.
r_2 = 0.5000 in.
r_3 = 0.5625 in.
r_4 = 0.625 in.
k_1 = 0.0717 BTU/hr-ft-°F, rubber hose
k_2 = 8.09 BTU/hr-ft-°F, steel braid
k_3 = 0.0717 BTU/hr-ft-°F, rubber hose
T_1 = Oil Temperature
T_2 = Ambient Air Temperature in Vehicle Cab
L = Hose Length

At the end of run #81 the oil temperature = 236°F, ambient air = 60°F, L = 30 ft. (from reservoir to DCV to winch and return loop). When analyzing dissipation of heat from the pipe, the average temperature of oil must be regarded for the hour under consideration. Therefore, T_2 = 199°F $((236_{\text{end temp}} + 162_{\text{start temp}})/2)$.

$$Q = \frac{2(30\ ft)(199^\circ F - 80^\circ F)}{\frac{1}{.0717}\ln(\frac{.5}{.4275}) + \frac{1}{8.09}\ln(\frac{.5625}{.5}) + \frac{1}{.0717}\ln(\frac{.625}{.5625})}$$

$$Q = \frac{7{,}140}{\frac{.157}{.0717} + \frac{.118}{8.09} + \frac{.104}{.0717}}$$

$$Q = 1{,}956\ BTU/hr.$$

$$T = \frac{1{,}956\ BTU/hr.}{718.2\ lb \times .509\ BTU/lb\text{-}^\circ F} \times 1\ hr.$$

$$T = -5.35^\circ F$$

g. Cooling in piping due to convection and radiation.

$$Q = (h_c + h_r)\ A\ (T_2 - T_1)$$

$h_c + h_r$ = 2.48 (Marc's Mechanical Engineering Handbook)[a]

$$Q = 2.48\ \text{BTU/hr-ft}^2\text{-°F} \times \frac{.5625}{12}\ \text{ft} \times 30\ \text{ft} \times (199 - 80)\text{°F}$$

Q = 415 BTU/hr.

T = Q/mc

$$T = \frac{415\ \text{BTU/hr}}{718.2\ \text{lb} \times .509\ \text{BTU/lb-°F}} \times 1\ \text{hr.}$$

T = **-1.14°F**

h. Cooling due to convection in reservoir. Each plate of the reservoir (top, bottom, and sides) must be analyzed to determine full conduction effect. For simplification of the calculation, the reservoir is assumed to be cubical.

For the vertical side plates $A_v = 24\ ft^2$

$(h_c + h_r)_v = 2.21$ (Marc's Mechanical Engineering Handbook)[b]

For the horizontal vertical plates facing up $A_u = 4\ ft^2$

$(h_c + h_r)_u = 2.47$ (Marc's Mechanical Engineering Handbook)[c]

For the horizontal vertical plates facing down $A_d = 4\ ft^2$

$(h_c + h_r)_d = 1.95$ (Marc's Mechanical Engineering Handbook)[d]

<u>Heat Loss</u> $Q = (h_c + h_r)A \times (T_2 - T_1)$

$$Q_v = 2.21\ BTU/hr\text{-}ft^2\text{-}^\circ F \times 24\ ft^2 \times (199 - 80)^\circ F = 6{,}312\ BTU/hr$$
$$Q_u = 2.47\ BTU/hr\text{-}ft^2\text{-}^\circ F \times 4\ ft^2 \times (199 - 80)^\circ F = 1{,}176\ BTU/hr$$
$$Q_d = 1.95\ BTU/hr\text{-}ft^2\text{-}^\circ F \times 4\ ft^2 \times (199 - 80)^\circ F = 928\ BTU/hr$$

$$\text{Total} = 8{,}461\ BTU/hr$$

$T = Q/mc$

$$T = \frac{8{,}461\ BTU/hr}{718.2\ lb. \times .509\ BTU/lb.\text{-}^\circ F} \times 1\ hr = \underline{\mathbf{-23.15^\circ F}}$$

II. Determination of Maximum Hydraulic Oil Heat Gain during Duty Cycle

SUMMARY

Based on maximum temperatures and resulting heat gains, an oil cooler capable of rejecting a minimum of 660 BTUs/min is required to maintain oil temperatures within the M88A1E1 parameters set by the vehicle's PD.

M88A1E1 Hydraulic oil system capacity:

All Hydraulic Hoses	8.34 gal
Spade Cylinder	5.87 gal
Boom & Stayline Cylinder	12.67 gal
Reservoir	75.00 gal
TOTAL	101.88 gal

Main Pump Maximum Flow Rate = 57 gal/min

Specific Gravity of Oil = .8

Specific Heat of Oil = .509 BTU/lb.-°F (taken at 176°F)

(Heat Transfer, by J.P. Holman)[e]

Density of Oil = 53.19 lb/ft^3

53.19 lb/ft^3 x .1337 ft^3/gal = 7.12 lb/gal

Mass of Oil in the System

101 gal x 7.12 lb/gal = 718.2 lb

BTU's Required to Change 101 gal 1°F

.509 BTU/lb-°F x 718.2 lb = 366 BTU/°F

Maximum Hourly Heat Gain - Based on One Hour of Winching for the Three Pump System.

Start at 162°F (170°F maximum allowable) in reservoir. This was the best data available to represent maximum operating conditions. Using runs 79 - 81 (time: 11:23 am - 12:23 pm) In the hour, three main winch and two aux winch operations were made.

For a one-hour period, the temperature rise = 236°F - 162°F = 74°F/hr

$$74°F/hr \times 366 \text{ BTU/°F} = 27{,}084 \text{ BTU/hr} = 451 \text{ BTU/min}$$

The 451 BTU/min represents the BTU's added to the hydraulic fluid on a 60°F day.

On an average day of duty cycle tests, a down-time occurs before the start of the next full cycle. During this down-time, the charge pump continued to run thus circulating the hydraulic fluid in the system under no load. This recirculating produces a cooling effect as heat is dissipated through the system. Test data taken indicates that the temperature dropped as much as 20°F in a 35 minute down-time interval. This 20°F is heat that has been added to the system during the prior work cycles and therefore must be considered. It must be added to the 451 BTU/min rate to approximate the maximum heat rejection.

$$20°F/35 \text{ min} = .57 \text{ °F/min}$$

$$.57°F/min \times 366 \text{ BTU/°F} = 209 \text{ BTU/min}$$

$$451 \text{ BTU/min} + 209 \text{ BTU/min} = \underline{660 \text{ BTU/min}}$$

The 660 BTU/min represents the total amount of heat being absorbed by the hydraulic oil. The maximum temperature allowable for the hydraulic oil to ensure safe system operation is 170°F. During continuous field operations in a hot environment, the hydraulic reservoir temperature will attain 170°F at some point in time. If continuous winching operations are made, it can be reasonably assumed that the hydraulic system will be gaining 660 BTU's/min as calculated above. This heat will need to be rejected to maintain the 170°F requirement. As a starting point, a hydraulic oil cooler should be installed in the M88A1E1 of a size sufficient to reject as a minimum 660 BTU's/min. Upon installation, further vehicle tests will be required to optimize the cooler size.

III. Preliminary Cooler Selection

A brief market research of hydraulic oil coolers was made. The review determined the Thermal Transfer Corporation of Racine, Wisconsin, to be a prime supplier of coolers to the hydraulics industry. Product data was obtained. Selection of a cooler size would be a typical example of a cooler that industry could supply.

The following three pages refer to product data for the Thermal Transfer "One-Pass" Oil Cooler. One pass refers to the oil only travelling in one direction while in the cooler.

One important design consideration is size. To install the cooler, it is necessary to design a ballistic housing to hold the cooler. The larger the cooler, the larger the housing. The cooler is to be mounted on the rear deck of the vehicle adjacent to the auxiliary power unit. As size increases, the possibility of interferences with vision and hydraulic operation increases. For the preliminary selection of coolers, the smallest cooler that could handle the load requirement was chosen.

Knowing the vehicle cooling requirements of 660 BTU/min and an oil flow of 21 gal/min, it was determined that an AO-25 oil cooler was required. The AO-25 has a range for cooling of 460-670 BTU/hr depending on oil flows and pressure drop across the cooler. A 660 BTU/min requirement is at the high end of the coolers capability. The AO-25 was selected to keep the sizing down. An AO-30 would be an alternate choice.

IV Theoretical Calculation of Oil Cooler Size

Taking the dimensions, material properties, and air/oil flow requirements of the Thermal Transfer AO-25, a theoretical calculation has been made to verify the sizing data supplied for the AO-25 cooler. As a result of the calculation, it has been determined that the cooler should reduce the oil temperature by a minimum of 584 BTU/min. The AO-25 range of cooling is or 460- 670 BTU/min. The 584 BTU/min theoretical value falls in the middle of this range, thus substantiating the design.

An AO-25 oil cooler supplies 2,240 cu ft/min air flow

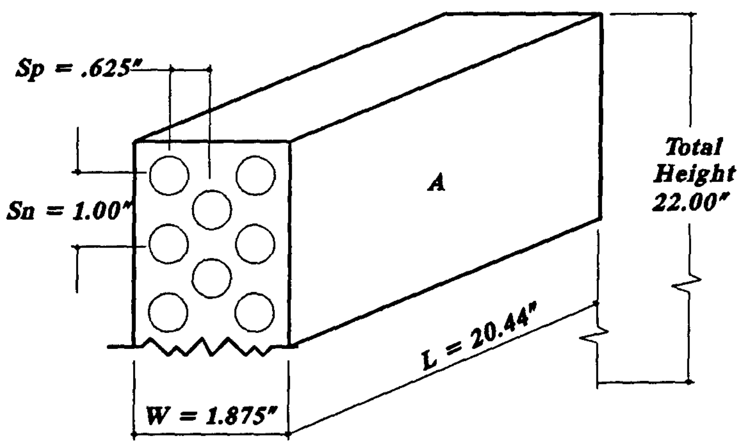
Air flow = 2,240 ft^3/min x 1 m^3/35.134 ft^3 x 1 min/60 sec

= 1.0626 m^3/sec

Cooler Dimensions:

Total of 61 tubes; outside tube diameter (d_o) = .396 in = .01005 m

Face Area = .5191 m x .5588 m = .29007 m^2



AO-25 Oil Cooler Dimensions

1. Calculation of Heat Transfer Coefficient at Air Film (h_o)

Assume: Ambient air at one atmosphere air at 120 °F (49°C). For the length of copper tubing, the external wall remains at 236°F (113.33°C).

The maximum temperature gained during testing (236°F) and the maximum outside design condition for ambient air (120°F) have been used in this calculation. It is the objective of this calculation to size a cooler that will operate under the extreme working conditions.

Air Properties at Film Temp (T_f):

$$T_f = \frac{T_w + T_\infty}{2} = \frac{113.3 + 49}{2} = 81.11°C = 354.2°K$$

$$\rho_f = \frac{P}{RT_f} = \frac{1.0132 \times 10^5}{287 \times 354.2} = .9965 \text{ kg/m}^3$$

ρ_f = Density at film temperature

μ_f = Dynamic viscosity = 2.093×10^{-5} Kg/m^3

k_f = Thermal conductivity = .03034 W/m·°C

c_p = Specific heat = 1.0094 kJ/Kg·°C

Pr = Prandtl number = .6963 (dimensionless)

Maximum Velocity of Air Through Tube Bank

$$\text{Air velocity } (v_\infty) = \frac{1.0626 \text{ m}^3\text{/sec}}{.29007 \text{ m}^2}$$

$$v_\infty = 3.6632 \text{ m/sec}$$

$$v_{max} = v_\infty \times \frac{s_n}{s_n - d_o}$$

$$v_{max} = 3.6632 \times \frac{1}{1 - .01005}$$

$$v_{max} = 3.7 \text{ m/sec.}$$

Heat Transfer Coefficient (h_o)

$$Re \text{ (Reynolds number)} = \frac{\rho V_{max} d}{\mu} = \frac{.9965 \times 3.7 \times .01005}{2.09 \times 10^{-5}}$$

$$Re = 1773$$

$$\text{Heat transfer coefficient} = \frac{hd_o}{k_f} = C(Re)^n \times Pr^{1/3}$$

$$\frac{s_p}{d_o} = \frac{.01588}{.01005} = 1.579$$

$$1'' = .0254\ m$$

$$\frac{s_n}{d_o} = \frac{.0254}{.01005} = 2.386$$

• For staggered tubes in cooler[f]

C = .526, n = .567

$$h_o = \frac{C \times Re^n \times Pr^{1/3} \times k_f}{d_o} = \frac{.526 \times (1773)^{.567} \times .696^{1/3} \times .0303}{.01005}$$

h_o = 97.74 W/m^2·°C for ten rows of fin tube

For three rows deep of staggered tubes[g], multiply by .83

h_o = .83 x 97.74 = 81.13 W/m^2·°C

h_o = 14.28 Btu/hr·ft^2·°F

2. Calculation of Oil Cooler Exit Air Temperature

Total tube surface area to be considered in the 61 tube bank

$$A = N \times \pi \times d_o \times L$$

$$A = 61 \times \pi \times .01005 \times .5588$$

$$A = 1.0771\ m^2$$

Air flow through the tube bank increases air temperature

∞ = Ambient Air

w = Wall

sub1 = Entrance

sub2 = Exit

Energy balance

$$h_o A\left(T_w - \frac{T_{\infty 1}+T_{\infty 2}}{2}\right) = \dot{m}c_p(T_{\infty 2}-T_{\infty 1})$$

49°C = 322.16°K

Mass flow rate of air into the cooler

$$\dot{m} = \rho_\infty v_\infty A$$

$$\rho_\infty = \frac{P}{RT} = \frac{1.0132 \times 10^5}{287 \times 322.16}$$

$$\rho_\infty = 1.0958 \text{ kg/m}^3$$

$$\dot{m} = 1.0958 \times 3.7 \times .29007$$

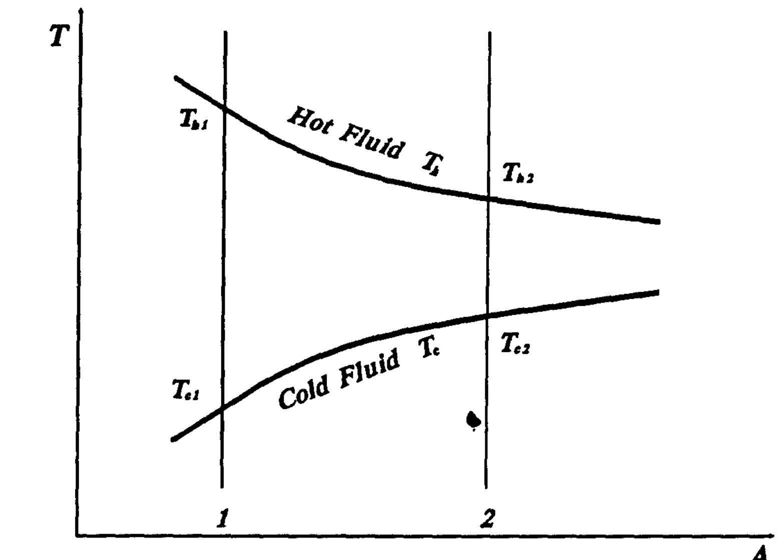
$$\dot{m} = 1.176 \text{ kg/sec}$$

Substituting into the energy balance equation

$$81.13 \times 1.07712 \times \left(113.33-\frac{49}{2}-\frac{T_{\infty 2}}{2}\right) = 1.176 \times 1009 \times (T_{\infty 2}-49)$$

$$T_{\infty 2} = 53.57°C$$

3. <u>Calculation of Log Mean Temperature Differential (ΔT_m)</u>



Temperature Profile in Heat Exchanger

T_{h1} = 113.33°C $\quad$ T_{h2} = 76.66°C
T_{c1} = 48.88°C $\quad$ T_{c2} = 53.57°C

$$\Delta T_m = \frac{(T_{h2}-T_{c2})-(T_{h1}-T_{c1})}{\ln\left[\frac{T_{h2}-T_{c2}}{T_{h1}-T_{c1}}\right]}$$

$$\Delta T_m = \frac{(76.66-53.57)-(113.33-48.88)}{\ln\left[\frac{76.66-53.57}{113.33-48.88}\right]}$$

ΔT_m = 40.29°C

4. <u>Calculation of Heat Transfer Coefficient for Oil in the Tubes (h_i)</u>

Copper tube O.D. $= d_o =$ .0100584 m

Copper Tube I.D. $= d_i =$ .00922 m

Properties of oil at 236°F (113.3°C)

$\rho =$ 832.643 kg/m^3

k = .13566 W/m·°C

$\mu = \gamma \times \rho$

$\gamma = .1503 \times 10^{-4}$

$\mu = .1503 \times 10^{-4} \times 832.63$

$\mu = 1.251 \times 10^{-2}$ kg/m·sec

$Pr = 208.66$

Mass Flow Rate of Oil

$$\dot{m} = \frac{21 \text{ gal}}{\text{min}} \times \frac{1 \text{ min}}{60 \text{ sec}} \times \frac{.00378 \text{ m}^3}{\text{gal}} = 1.323 \times 10^{-3} \text{ m}^3/\text{sec}$$

Assume a one pass oil flow in oil cooler for 61 tubes

$$\dot{m} = \frac{1.323 \times 10^{-3} \text{m}^3/\text{sec}}{61 \text{ tubes}}$$

$$\dot{m} = 2.16885 \times 10^{-5} \text{ m}^3/\text{sec per tube}$$

Velocity of oil in tube

$$v = \dot{m}/A_i$$

$$v = 2.16885 \times 10^{-5} \text{ m}^3/\text{sec} \times \frac{1}{6.6768 \times 10^{-5} \text{ m}^2}$$

$$v = .3248 \text{ m/sec}$$

$$Re = \frac{\rho v d_i}{\mu} = \frac{832.64 \times .3248 \times .00922}{1.251 \times 10^{-2}}$$

$$Re = 199.33$$

Turbulent flow in tubes occurs when: $10 < L/d_i < 400$

$L/d_i = .5588/.00922 = 60.60$, therefore turbulent flow

For turbulent flow, the Nusselt number equals

$$N_u = .036 Re^{.8} Pr^{.33} (d/L)^{.055}$$

$$N_u = .036 \times (199.33)^{.8} \times (208.66)^{.33} \times (.00922/.5588)^{.055}$$

$N_u = 11.57$

Solving for the heat transfer coefficient of oil

$h_i = N_u(k/d_i) = 11.78(.13567/.00922)$

$h_i = 170.24\ W/m^2 \cdot °C$

$h_i = 29.98\ BTU/h \cdot ft^2 \cdot °F$

5. Calculation of Overall Heat Transfer Coefficient (U)

$$U = \frac{1}{\left(\frac{A_o}{A_i}\right)\left(\frac{1}{h_i}\right) + \frac{A_o \ln(r_o/r_i)}{2\pi kL} + \frac{1}{h_o}}$$

$A_o = d_o \pi L = .01005m \times \pi \times .5588m = .01764\ m^2$

$A_i = d_i \pi L = .00922m \times \pi \times .5588m = .01618\ m^2$

For copper, $k = 373\ W/m \cdot °C$

$$U = \frac{1}{\left(\frac{.01764}{.01618}\right)\left(\frac{1}{173.34}\right) + \frac{.01764 \ln\left(\frac{.005025}{.00461}\right)}{2\pi(373).5588} + \frac{1}{81.13}}$$

$U = 53.39\ W/m^2 \cdot °C$

6. Overall Heat Transfer for Tubing

$q = UA\Delta T_m$

$q = 53.72\ (61 \times \pi \times .01005 \times .5588)\ 40.29$

$q = 2{,}315.04\ W$

$q = 7{,}899.16\ BTU/hr$

$q =$ **132 BTU/min**

Having calculated the heat rejection for the cooler tubing, it is now necessary to calculate the heat rejected by the cooler fins. The fin element is the main source of heat rejection. The following geometries were analyzed to arrive at the overall heat rejection rate.

7. Convective Heat Transfer Coefficient for One Fin

   Using the properties of air given in Section 1., the average heat transfer coefficient (h) is calculated. The geometry of one fin is defined as being 1.875" (.04762 m) wide, 22" (.5588 m) long, and .010" thick. Additionally, 61 holes for the tubes having a diameter of .396" (.01005 m) are considered negative area.

$$\frac{hL}{k} = Nu = .664\ Re^{.5}\ Pr^{.33}$$

$$Re = \frac{\rho v_{max} w}{\mu}$$

$$Re = \frac{.9965 \times 3.7 \times .04762}{2.09 \times 10^{-5}}$$

$$Re = 8401.71$$

$$Nu = .664 \times 8401.71^{.5} \times .6963$$

$$Nu = 53.94$$

$$h = \frac{Nu\ k}{w}$$

$$h = \frac{53.94 \times .03034}{.04762}$$

$$h = 34.37\ W/m^2 \cdot {}^{o}C$$

8. Overall Heat Transfer for Fins

$$q = hA(T_w - T_\infty)$$

$$A = L \times w - (n \times \frac{\pi d^2}{4})$$

$$A = .5588 \times .04762 - (61 \times \frac{\pi \times .01005^2}{4})$$

$$A = .02176\ m^2$$

q = 34.37 x .02176 (113.33 - 48.88)

q = 48.21 W

q = 2.742 BTU/min (for one fin)

For all the fins in the cooler multiply by 165

q = 452 BTU/min

9. Total Heat Reduction for the Cooler (tubing and fins)

Tubing	=	132 BTU/min
Fins	=	452 BTU/min

584 BTU/min Total Heat Rejection by the Cooler

Appendix K List of References

[a] Baumeister, Theodore, and Marks, Lionel S., "Standard Handbook for Mechanical Engineers," McGraw-Hill Book Co., NY, p. 4-106 (1967)

[b] Baumeister, Theodore, and Marks, Lionel S., "Standard Handbook for Mechanical Engineers," McGraw-Hill Book Co., NY, p. 4-106 (1967)

[c] Baumeister, Theodore, and Marks, Lionel S., "Standard Handbook for Mechanical Engineers," McGraw-Hill Book Co., NY, p. 4-106 (1967)

[d] Baumeister, Theodore, and Marks, Lionel S., "Standard Handbook for Mechanical Engineers," McGraw-Hill Book Co., NY, p. 4-106 (1967)

[e] Holman, J.P., "Heat Transfer," McGraw-Hill Book Co., NY, p. 641 (1986)

[f] Holman, J.P., "Heat Transfer," McGraw-Hill Book Co., NY, p. 300 (1986)

[g] Holman, J.P., "Heat Transfer," McGraw-Hill Book Co., NY, p. 300 (1986)

Definitions

gpm	=	gallons per minute
psi	=	pounds per square inch
hp	=	horsepower
hp	=	gpm x psi x .000583 (hydraulic horsepower)
gal	=	gallons
m	=	mass
m	=	meters
V	=	volume
ρ	=	density
Q	=	heat gain
q	=	heat gain
BTU	=	British Thermal Unit
hr	=	hour
T	=	temperature
c_p	=	specific heat
L	=	length
ft	=	feet
ΔT	=	change in temperature
k	=	thermal conductivity
r	=	radius
in	=	inches
ln	=	natural log
h	=	heat transfer coefficient
h_r	=	radiation heat transfer coefficient
h_c	=	conduction, convection heat transfer coefficient
A	=	Area
v_∞	=	incoming air velocity
T_f	=	film temperature of air
T_∞	=	ambient air temperature
T_w	=	tube wall temperature
P	=	pressure
R	=	gas constant of air
μ	=	dynamic viscosity
s	=	distance
$\dot{m}$	=	mass flow rate
γ	=	kinematic viscosity
Re	=	Reynolds Number
Pr	=	Prandtl Number
Nu	=	Nussult Number
t	=	thickness

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